REPORT No. 55

INVESTIGATION OF THE MUFFLING PROBLEM FOR AIRPLANE ENGINES

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

PREPRINT FROM FIFTH ANNUAL REPORT

WASHINGTON
GOVERNMENT PRINTING OFFICE
1920

3195
10

REPORT No. 55

INVESTIGATION OF THE MUFFLING PROBLEM FOR AIRPLANE ENGINES

BY

G. B. UPTON and V. R. GAGE

REPORT No. 55.

INVESTIGATION OF THE MUFFLING PROBLEM FOR AIRPLANE ENGINES.

By G. B. Upton and V. R. Gage.

The initial perception of the presence of an airplane comes commonly through hearing rather than sight. When near a plane the noise of the unmuffled engine is fairly deafening. If muffling can be contrived without too large a loss of power it will become much easier for the pilot to operate his plane, a cut-out being provided for engine-testing purposes. In civil use of planes, if passengers are to be carried or if planes become numerous, muffling will almost surely be required, as it now is for automobiles, motor boats, and stationary engines. In military use of the planes the advantages to be derived from silent operation, if that is possible, are immensely greater; for example, with night-bombing airplanes.

A preliminary report upon this subject was printed as Report No. 10 of the Second Annual Report of the National Advisory Committee for Aeronautics, 1916, pages 41 to 49; hereinafter referred to as Report No. 10. This report outlined the problem, gave the status of muffling for automobile engines, and gave the beginnings of our experimental work. For the main part of the experimental work, Prof. V. R. Gage has been associated with the initial staff of Profs. H. Diederichs and G. B. Upton.

In the early summer of 1917 the Curtiss engine was taken over by the U. S. A. S. M. A. at Ithaca. A considerable amount of experimental data had already been accumulated. Designs of mufflers had been worked out to be tried upon planes in field work. At this time, however, muffling was much less important than the production of engines and planes, so that field experiments with mufflers were not carried out.

The work has fallen into two divisions: First, the determination of the relation between back pressure in the exhaust line and consequent power loss, for various combinations of speed and throttle positions of the engine. Second, the construction and trial of muffler designs, covering both type and size. The main body of the work has been done on a Curtiss OX eight-cylinder airplane engine, 4 by 5 inches, rated 70 horsepower at 1,200 revolutions per minute. For estimation of the muffling ability and suppression of "bark" of individual exhausts, we have also used an "Ingeco" stationary, single cylinder, 5½ by 10 inch, throttling governed gasoline engine, and occasionally other engines.

On the Curtiss engine the carburetor was a Schebler model L. The throttle control rod was graduated and adapted for duplication of settings by means of a screw setting into holes. Adjustment of needle valve and cams controlling the mixture was once made and was not subsequently changed. The ignition was by Bosch magneto with fixed spark, set in the advanced position by the manufacturer's instructions. This adjustment was never changed. The spark plugs gave trouble and had to be renewed. The engine was started by power from an electric (street car) motor, belted to a pulley which was keyed on the end of the fan dynamometer shaft, on the end away from the engine. The engine was brought to a moderate speed with the magneto short circuited and then ignition was turned on and the belt thrown off the fan dynamometer pulley simultaneously. Cooling water was supplied to the engine from the water mains through the regular circulating pump. The water supply valve was always opened to the same point, which had been found by trial to give slightly more than adequate cooling. It was found that air locks might occur and to detect them separate discharges lines were used, one from each block of cylinders.

In the determination of power losses due to the mufflers it was desirable that the engine should drive a dynamometer with the same torque-speed characteristics as a propeller. We therefore built a fan dynamometer, copying in detail the dimensions of one at the Automobile Club of America's laboratory at New York City. This design of fan has previously twice been calibrated using cradle dynamometers. The fan has two blades set diametrically opposite. These blades are rectangular plates 14 by 10 inches, with the 10-inch dimension radial. As used, the outside diameter across the blades was 42 inches, requiring 35.4 horsepower at 1,000 revolutions per minute, and for other powers varying with the cube of the speed. This adjustment holds the engine to rated load at rated speed (61.2 horsepower at 1,200 revolutions per minute, 69.1 horsepower at 1,250, 77.8 horsepower at 1,300). Dimensions and calibration of the fan are given more completely in Appendix A.

The determination of the power consumed by the fan depends solely upon the measurement of speed. For the reading of speed we used a Hopkins electrical tachometer, with its dynamo driven directly by the engine crankshaft through a flexible coupling. The tachometer was calibrated several times during the progress of the work, both in place on the engine and on a small high-speed lathe.

To determine the power loss due to back pressure, we put exhaust manifolds on the engine along each block of four cylinders and combined the exhausts in a cross pipe at the rear of the engine, as illustrated in figure 1, a photograph of the set-up. The cross pipe ended in a tee with two valves, one a gate valve for adjustment and the other a quick-opening valve, similar to a "molasses" valve. The second valve was used either entirely open or entirely shut; the change from closed to open and reverse could be made instantly. In running to determine power loss the engine would be set at a given throttle position with the quick-opening valve wide open; then closing that valve the gate valve was set to give a desired back pressure. Runs were then made, alternately, in quick succession, with the quick-opening valve open and shut. In each condition readings were taken of speed and back pressure. The alternation of conditions was repeated several times, until the drop of speed (and power losses) associated with a certain back pressure seemed well determined.

There was one manifold for each block of four cylinders composing one leg of the vee. The back pressures of the two groups of cylinders were read separately and independently. The manometer connections were made in each manifold at a point about 6 inches beyond the last cylinder and the two fittings were identical in construction. The pressure taps were made of small brass rods riveted over inside nearly flush with the manifold, with a lock nut outside the manifold. The openings through the rods were the same and were about $\frac{1}{16}$ inch in diameter. This construction was used in order to minimize as far as possible any errors of pressure due to the effect of velocity and to damp the pulsations of pressure. If any such velocity effects did enter into the observations of pressure they would disappear in the taking of the pressure difference of runs with and without applied back pressure. Taking the pressure readings so close to the engine was intentional; all manifold and piping resistance beyond these taps will be shown in the readings, and it was a matter of interest to discover how serious these losses might be, due to the use of long or improper exhaust piping.

From the pressure taps connection was made through rubber tubing of about $\frac{1}{4}$-inch inside diameter and 6 feet long to mercury manometers. A considerable mass of mercury was used in each manometer to obtain the damping effect of its inertia. Further damping was found necessary and was secured by stuffing the upper ends of the manometer tubes with cotton waste. With the engine running, the back-pressure readings were satisfactorily steady and practically without time lag in shifting from running with and without applied back pressures.

The first series of runs was made to determine the power losses caused by different applied back pressures with the engine under a fixed throttle position, set for normal power and speed. The observations taken are shown in the first four columns of Table I. The other values shown in Table I are the results of computations from the averages of the data.

The true speed, revolutions per minute, is obtained from the tachometer readings by the use of calibrations made before and after groups of tests. The correction factor for the tacho-

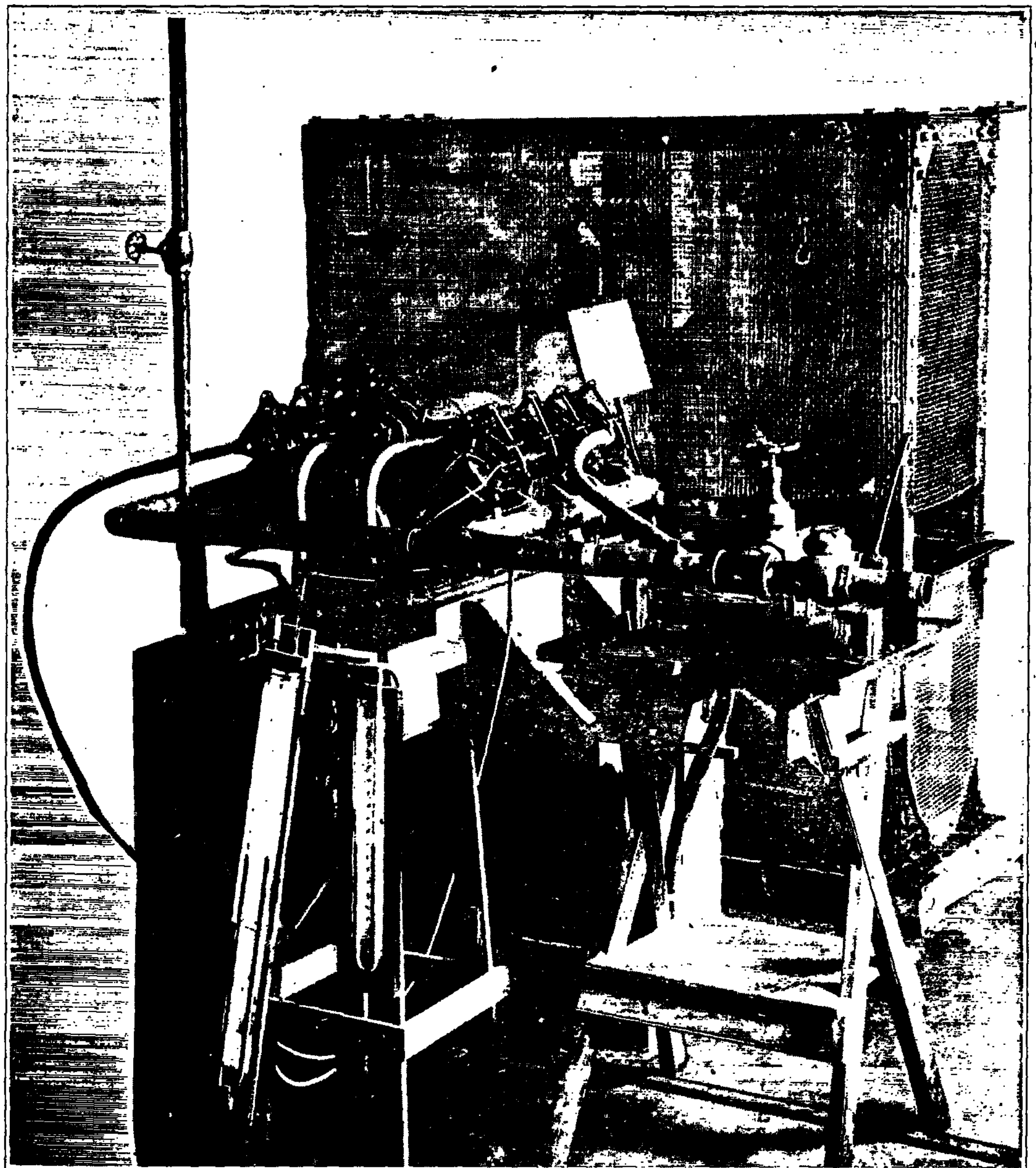


FIG. 1.

meter was always a constant multiplied into the indicated speed. This constant was about 1.09.

The per cent power loss was computed from the relative speeds with and without applied back pressure. If the speed with applied back pressure was 97 per cent of the original speed, the power loss was $(1.00)^3-(0.97)^3=1.00-0.9127=8.7$ per cent. This relation is true because of the loading of the engine by the fan dynamometer. The power absorbed by the fan is proportional to the cube of the speed. (Appendix A.)

The brake M. E. P. (brake mean effective pressure) is the product of the mechanical efficiency of the engine and the mean effective pressure as shown by an indicator diagram, and is expressed in pounds per square inch. For this engine:

Let P = brake M. E. P.
R = r. p. m.
B = cylinder bore diameter, in inches = 4 inches.
S = cylinder stroke, in inches = 5 inches.
N = number of cylinders = 8.
HP = brake horsepower.

Then

$$HP=\frac{P\times\frac{\pi}{4}B^2\times\frac{S}{12}\times\frac{R}{2}\times N}{33000}=\left(\frac{\pi B^2 SN}{4\times2\times12\times33000}\right)PR$$

$$=\left(\frac{1}{1577}\right)PR$$

$$P=\frac{(1577)}{R}\times HP.$$

For this fan dynamometer:

$$HP=KR^3$$

in which K is a constant whose value was 35.4×10^{-9} at the setting used. (Appendix A.)

Combining the expressions for this engine and fan:

$$P=\frac{1577}{R}\times KR^3=1577\times KR^2$$

$$=55.8\ R^2\times10^{-6}$$

Various results from this group of tests, as shown in Table I, are shown as curves in plots 2 to 5, inclusive. In all of these curves the abscissa is the applied back pressure measured in inches of mercury. This "applied back pressure" is strictly the increase of manometer reading, over that given with open discharge from exhaust manifolds and piping, with the application of a definite constriction of discharge.

Plot 2 shows the power losses, in per cent, as a function of the applied back pressure. With no applied back pressure the throttle was set and locked to give about 1,230 r. p. m. corresponding to about 65 horsepower output to the dynamometer. The engine did not always come back to this speed and power when the back pressure was relieved, because of many minor variations of ignition, carburetion, lubrication, cooling, etc. These changes were cared for by the method of testing employed, as previously described. The results as to power loss in plot 2 will be found only for one setting of the throttle on this engine. It could not be assumed that the same percentage power loss would be found, for a given applied back pressure, on another engine, or even with other throttle positions on this engine. These power losses are conditioned, also, by the dynamometer characteristic of power varying with cube of speed. They would not hold for automobile engine operation in general, because the load characteristics would be different. This curve is, however, approximately typical, in form and in numerical value, of average flying conditions with airplane motors, when the exhaust is choked by any means to increase the back pressure. This curve gives the information desired concerning the relative magnitude of the power losses incident to increasing the back pressure of an airplane engine by attempts at muffling. If satisfactory noise suppression can be secured with small increases of back pressure, the power loss may be tolerable.

For moderate back pressures the power loss is substantially proportional to the back pressure. For higher back pressures the power loss mounts rapidly, apparently at such a rate that a back pressure of even less than 10 pounds per square inch (20 inches mercury) would stop the engine. These relations are perhaps better brought out in plot 3, which is the same as plot 2 except that the coordinates are logarithmically scaled. The full line curve in plot 3 is the curve of plot 2 transferred. The dash line in plot 3 shows what would happen if the power loss continued to be proportional to the back pressure.

A possible explanation of this changing effect of back pressure as the back pressure increases may be found by considering the indicator card. This is schematically shown in figure 4. For small back pressures we may expect the main effect to be a lifting of the exhaust line of the card by an amount substantially equal to the increase of back pressure. The result would be a loss of indicated M. E. P. equal to the back pressure, because the elevation of the exhaust line would extend through the whole stroke. The loss of brake M. E. P. will be smaller than the loss of indicated M. E. P. in the ratio of the mechanical efficiency of the engine to unity.

At higher back pressures the exhaust gases are held back in greater amounts in the cylinder, leaving the clearance space, at the end of the exhaust period, filled with an adnormal weight of hot, dead gases. These, reexpanding, interfere with the incoming charge in various ways, lessening the amount of the fuel mixture drawn in. The decrease of charge quantity will result in a decrease of M. E. P. which is added to the decrease of M. E. P. due to lifting of the pressure of the exhaust line.

Probably it is the decrease of charge which is the principal reason for the possibility of stalling the engine by fairly completely choking the exhaust pipe and before complete closure is reached.

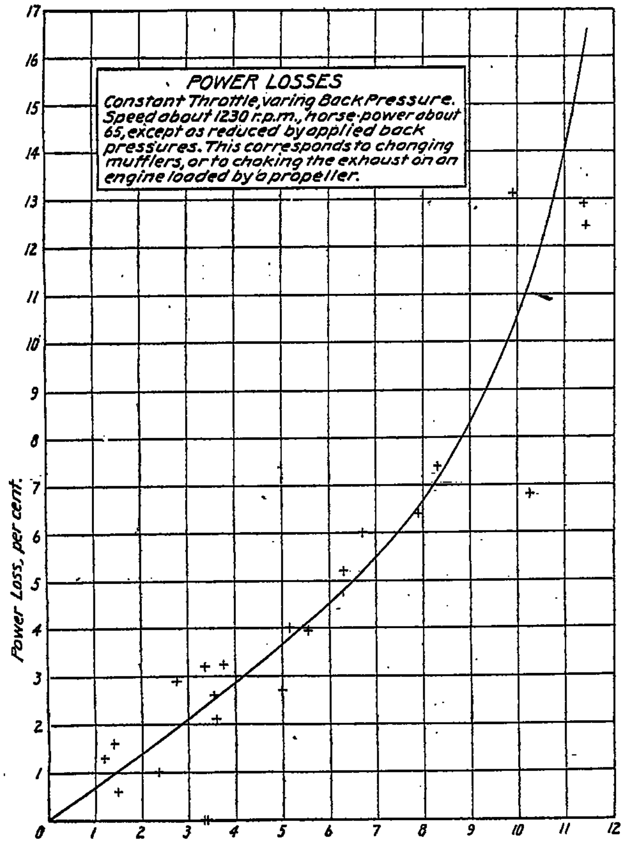
To make the findings of power loss caused by applied back pressure more general, not so much a matter of the particular case studied, the results are given as loss of brake M. E. P., by the curve on plot 5. The previous discussion of the effect of back pressure upon the indicator card explains the form of this curve and the relations between the test curve and the line of equality of brake M. E. P. loss and back-pressure increase. *For the smaller and reasonable values of back pressure it is quite safe to assume that the loss of brake M. E. P. (pounds per square inch) does not exceed, and is nearly equal to, the applied back pressure (in pounds per square inch).* This conclusion is of considerable importance to the designer of engines, exhaust manifolds, and mufflers, and is probably valid for all types and services of internal-combustion engines.

Of interest to the airplane designer is the loss of propeller speed consequent upon back pressure in the engine exhaust. Since propeller speed is tied, in a definite relation, to propeller power, the curve of plot 6 is really another version of plot 2. This curve of plot 6 should correspond approximately to average running conditions of airplane motors. It is about a three-quarter throttle position curve for this engine, and full throttle would probably change the form of the curve only at the higher back pressures.

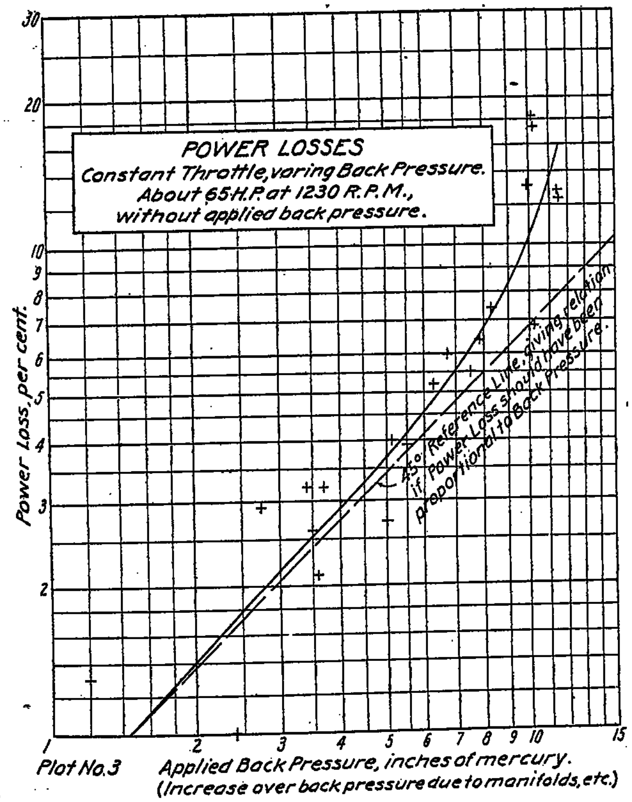
In the second series of tests the engine was brought up to about normal speed and power. The regulating valve at the end of the exhaust pipe was closed, allowing all the exhaust to leak through the joints of the piping and through the walls of the flexible metallic exhaust hose when the end outlet valve was closed. The value of back pressure was purposely made large, in order to give more accurate readings of back pressure, and to see what would happen. Readings of speed and back pressure were then taken with the end outlet valve alternately open and shut, until the values seemed to check among themselves. Then the throttle opening was reduced, and again readings were taken. Immediately after the smallest advisable throttle run, the normal throttle as used at first was reproduced, and a check run was made.

Table II gives the data and the computations of the runs made on August 24 and 29. The exhaust piping was changed between August 24 and 29 by making the pipe joints tighter.

Plot 7 shows the relation between the brake horsepower and the actual back pressure, as given in Table II. The back pressure increases as some exponential function of the horsepower,



Plot No. 2.



Plot No. 3

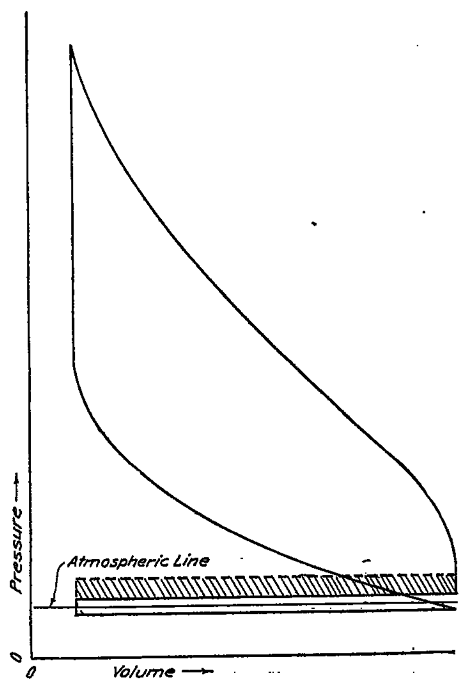
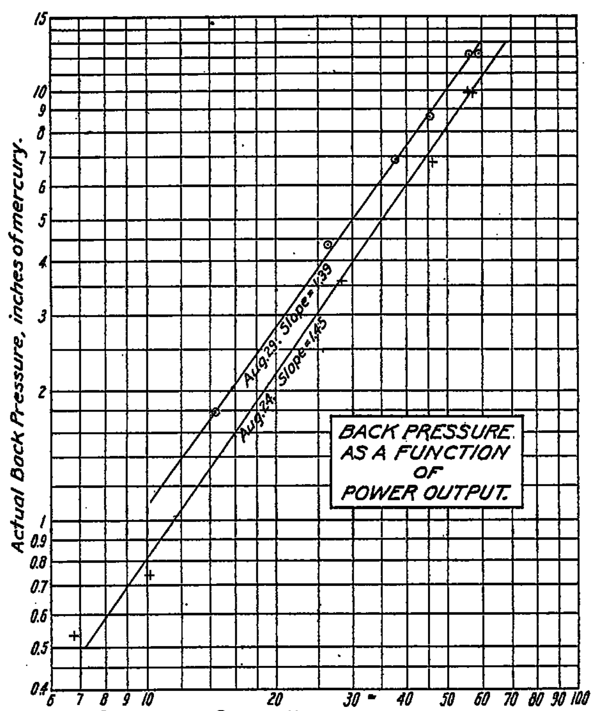
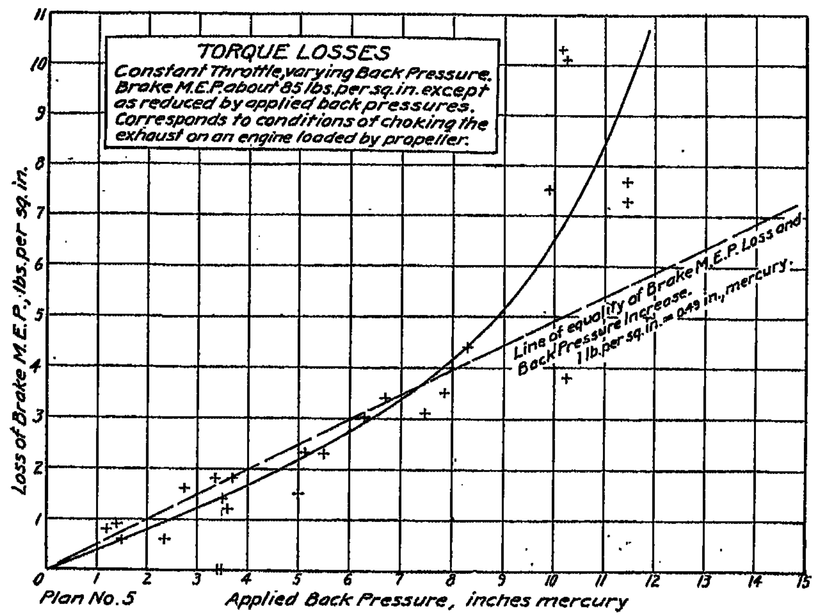
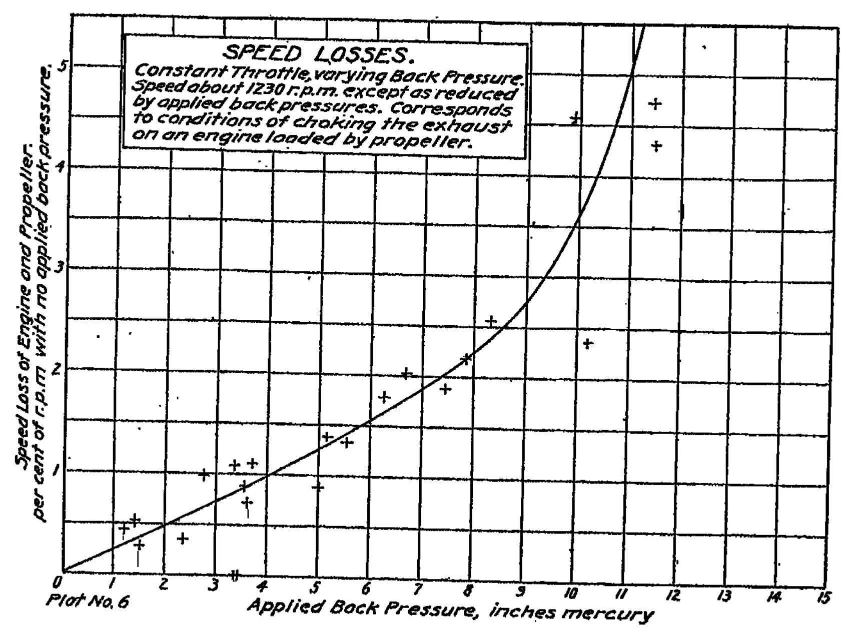


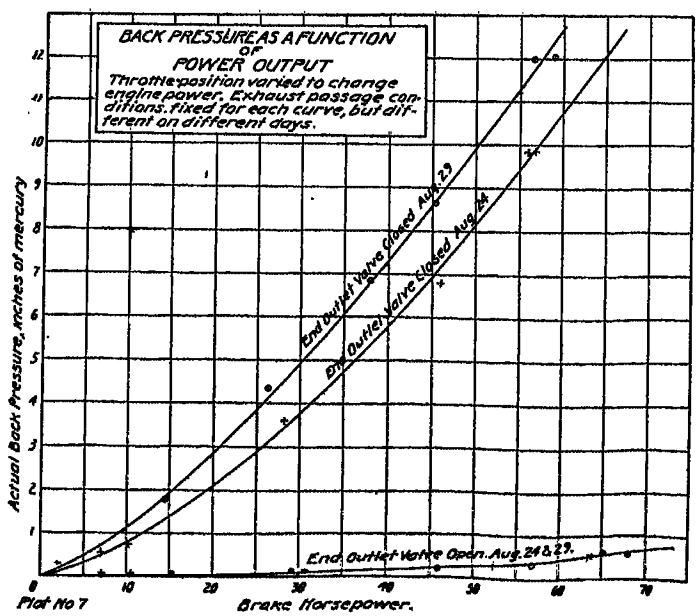
FIG. 4.

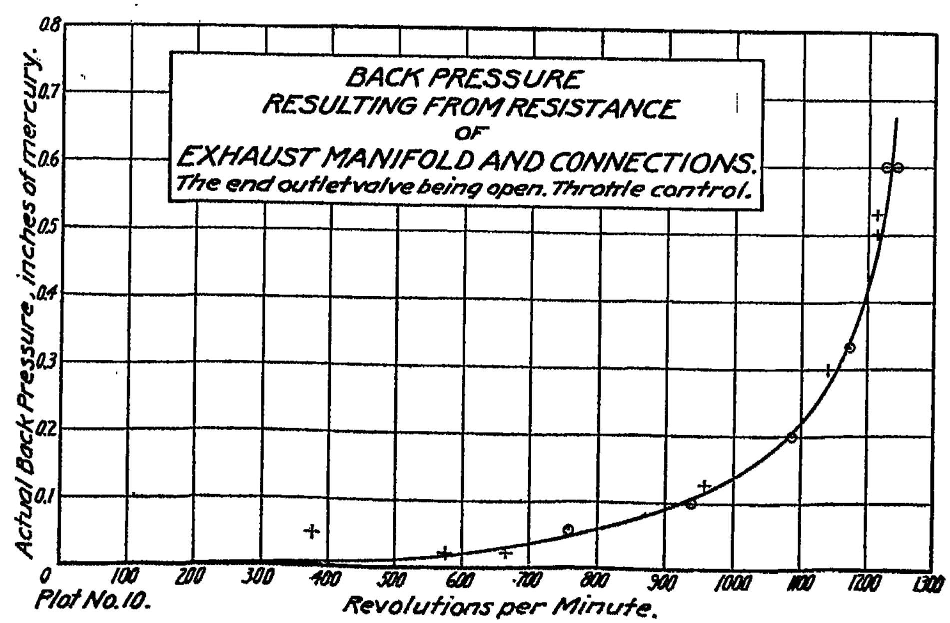


Plot No. 8


TORQUE LOSSES
Constant Throttle, varying Back Pressure. Brake M.E.P. about 85 lbs. per sq. in. except as reduced by applied back pressures. Corresponds to conditions of choking the exhaust on an engine loaded by propeller.
Line of equality of Brake M.E.P. Loss and Back Pressure Increase. 1 lb. per sq. in. = 2.049 in. mercury.
Loss of Brake M.E.P., lbs. per sq. in.
Applied Back Pressure, inches mercury
Plan No. 5


SPEED LOSSES.
Constant Throttle, varying Back Pressure. Speed about 1230 r.p.m. except as reduced by applied back pressures. Corresponds to conditions of choking the exhaust on an engine loaded by propeller.
Speed Loss of Engine and Propeller, per cent of r.p.m. with no applied back pressure.
Applied Back Pressure, inches mercury
Plot No. 6


BACK PRESSURE AS A FUNCTION OF POWER OUTPUT
Throttle position varied to change engine power. Exhaust passage conditions fixed for each curve, but different on different days.
End outlet Valve Closed Aug. 29
End Outlet Valve Closed Aug. 24
End Outlet Valve Open. Aug. 24 & 29.
Actual Back Pressure, inches of mercury
Brake Horsepower.
Plot No 7


BACK PRESSURE
RESULTING FROM RESISTANCE
OF
EXHAUST MANIFOLD AND CONNECTIONS.
The end outlet valve being open. Throttle control.
Actual Back Pressure, inches of mercury.
Revolutions per Minute.
Plot No. 10.

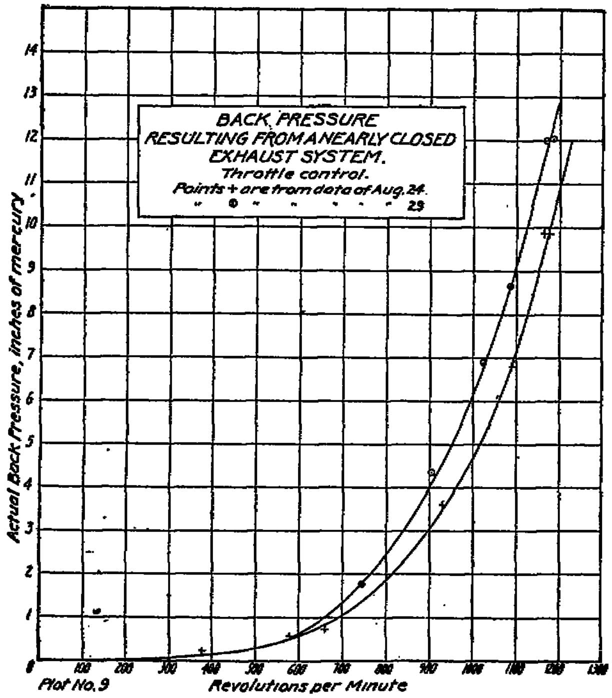
when the conditions of the exhaust passages remain unchanged. The same curves are shown on plot 8, using logarithmic instead of arithmetic coordinates. The slope of these curves is slightly less than 1.5, indicating about the 1.5 power of the engine output. An analysis of the muffler tests made at the University of Michigan (printed in Horseless Age, May, 1915) also indicates that the back pressure varies with about the 1.5 power of the engine output. (An analysis of some of the features of the University of Michigan report is given as Appendix B.)

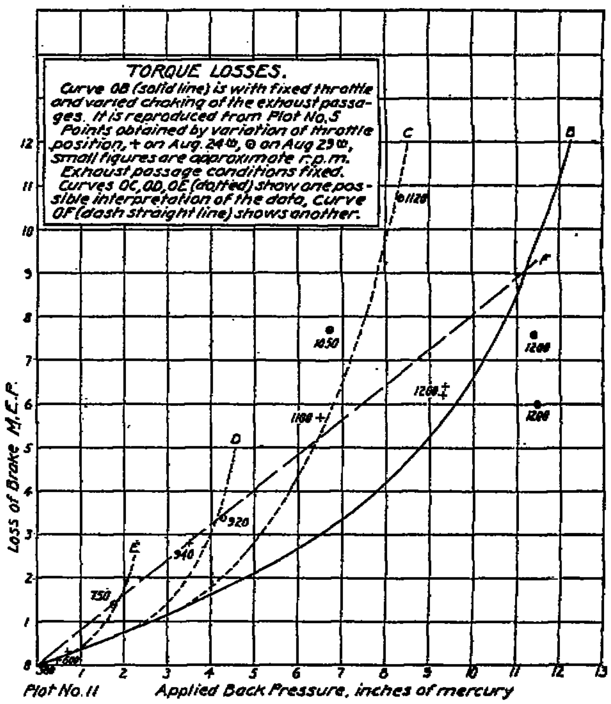
Neglecting changes of mechanical and thermal efficiency, the weight of exhaust gases, per unit of time, must be nearly proportional to the brake horsepower. So, for a rough approximation, the quantity of exhaust gas can be substituted for the brake horsepower, as abscissæ of the curves, plots 7 and 8. The quantity of fluid discharged through orifices is nearly proportional to the square root of the pressure difference across the orifices. By analogy it would be expected that the slope of the curves on plot 8 should be nearly 2. A contributing feature in the fact that they are not may possibly be explained by the common method of averaging and reading a fluctuating pressure. Averaging in case of pulsations is quite often done by damping. The damping of the oscillations may or it may not give a nearly true average of the pressure. But the instantaneous rate of flow through the orifice is proportional to the square root of the pressure at that instant. So for a pulsating flow there should be found the integration of the product of the average of the square roots of the pressures and the time interval during which the pressures existed, not the product of the square root of the average of the pressures and the total time. The latter method was used, agreeably to the common practice under such circumstances. The flow and the pressure difference in the exhaust manifolds are widely and rapidly pulsating. The average pressure as indicated on the manometer. may be far from the average pressure value which should be used in determining the flow However, when measuring the exhaust back pressures in this conventional manner, the empirical relation here shown may be quite useful—that the back pressure varies approximately as the horsepower of the engine, raised to the 1.5 power.

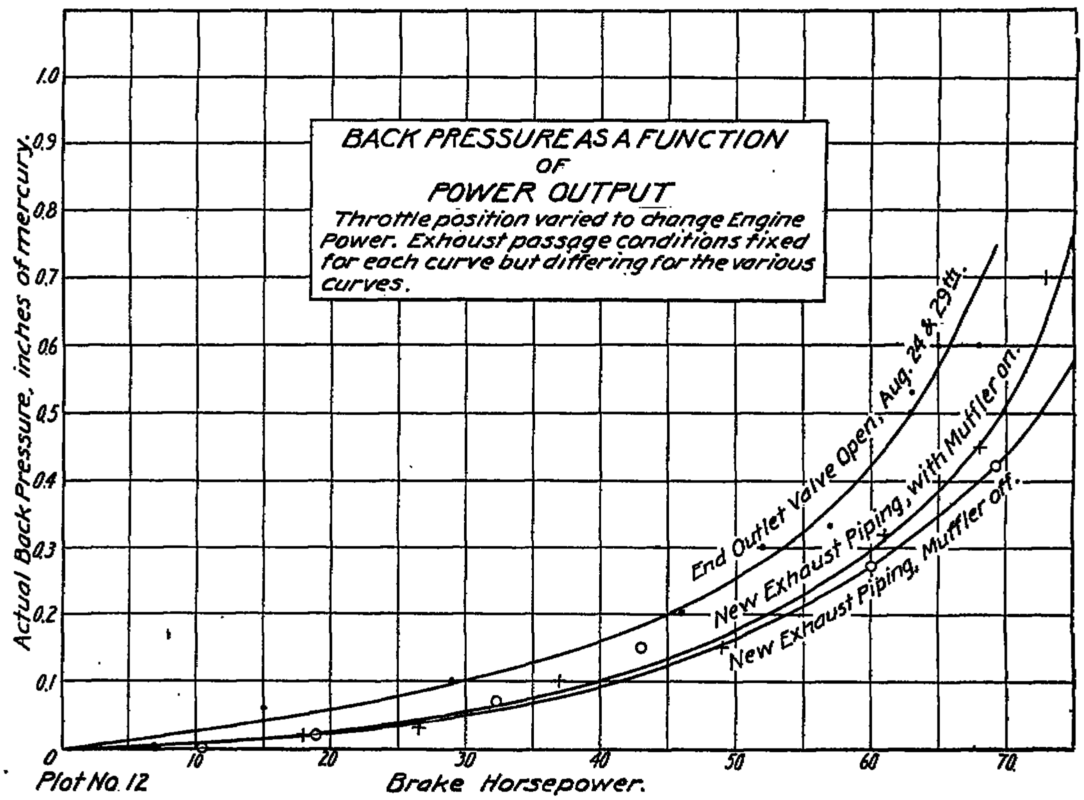
Plots 9 and 10 give information very applicable to muffling of airplane engines. They show how rapidly the back pressure rises to high values as the engine is opened out to full power and speed. The numerical values are obtained from Table II. Plot 9 shows the result of excessive choking of the exhaust which was purposely done in this case to bring out the action. The two curves of plot 9 result from slightly different initial conditions. Plot 10 shows the back pressure due to exhaust piping, only, including sharp turns, some extra pipe resistance, and fittings on our experimental set-up. This curve is of the same type and of the same magnitude of back pressure values which would come from a good muffler. It should be carried in mind that with these figures the engine was driving a fan whose characteristics correspond to an airplane propeller. The form of the curves is conditioned by this type of loading.

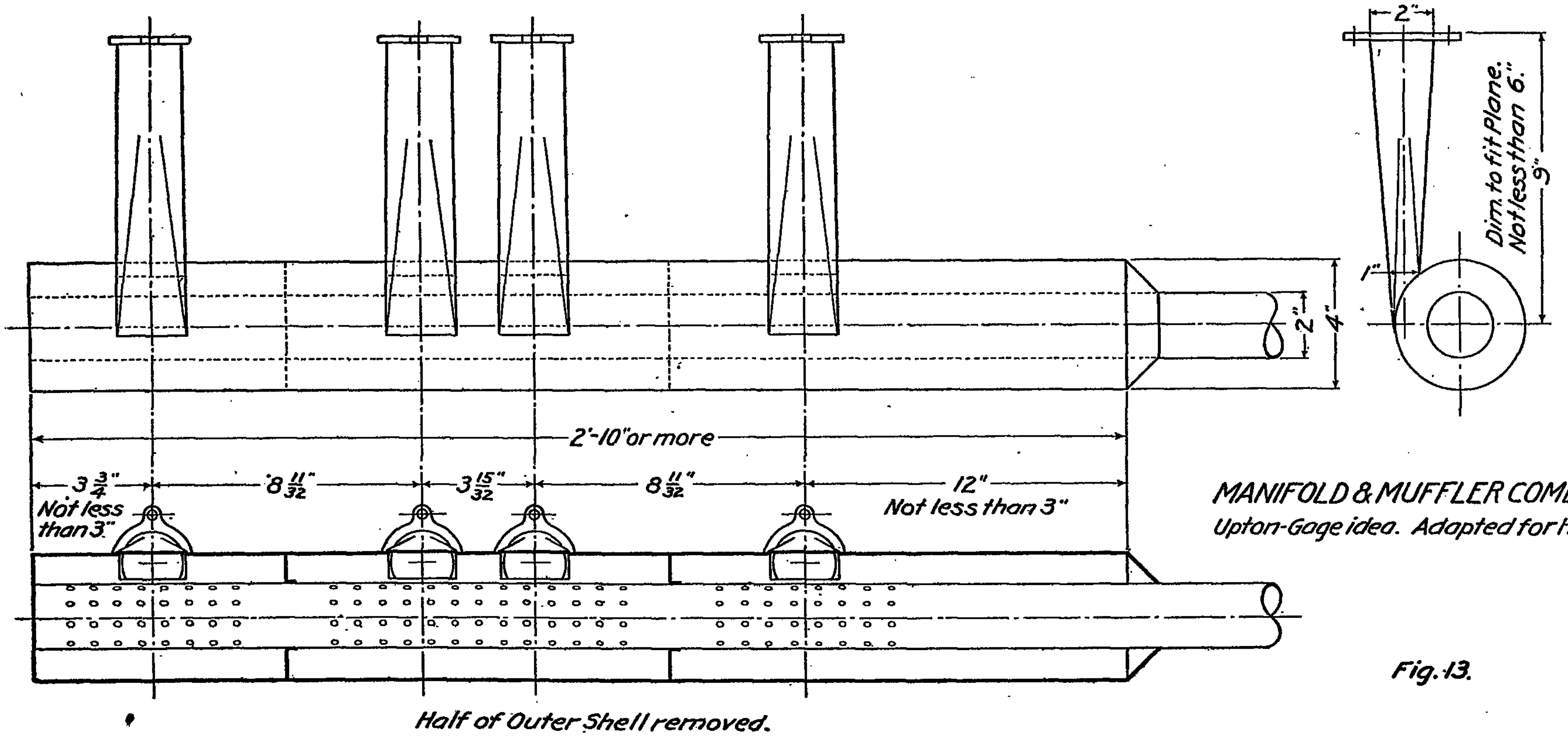
In attempting to analyze the data of Table II, reducing the loss of engine torque to terms of brake M. E. P., the result is the "shotgun" diagram of plot 11. Interpretation of this is highly speculative considering the small amount of data which was collected. The analysis here attempted is much broader than the special problem which was being studied, and so no special runs were made to get the information needed to complete this figure. Curve OB is transposed bodily from plot 5, representing runs at fixed throttle and varying exhaust conditions. The straight line OF on plot 11 represents, as near as possible, the average relation from the runs with varying throttle against fixed exhaust conditions. Since the engine was coupled to a fan dynamometer, speed and throttle position are tied together except for the slight modifications due to back pressures, etc. The small numbers adjacent to the points in plot 11 give the approximate r. p. m. of the engine and fan; and equal speeds nearly coincide with equal throttle positions. The apparent scattering of the points may indicate that the points do not belong upon one curve, but upon a family of constant throttle curves such as OC, OD and OE, all similar to OB. The curve OB was at about three-fourths throttle of the engine.

Table III and plot 12 are put in as a demonstration that the choking of the exhaust by sharp turns, pipe fittings, etc., give the same results as choking by a muffler. On August 29,


BACK PRESSURE
RESULTING FROM A NEARLY CLOSED
EXHAUST SYSTEM.
Throttle control.
Points + are from data of Aug. 24.
" ⊙ " " " " 29
Actual Back Pressure, inches of mercury
Revolutions per Minute
Plot No. 9


TORQUE LOSSES.
Curve OB (solid line) is with fixed throttle and varied choking of the exhaust passages. It is reproduced from Plot No. 5
Points obtained by variation of throttle position, + on Aug. 24th, ⊙ on Aug 29th, small figures are approximate r.p.m.
Exhaust passage conditions fixed.
Curves OC, OD, OE (dotted) show one possible interpretation of the data, Curve OF (dash straight line) shows another.
Loss of Brake M.E.P.
Applied Back Pressure, inches of mercury
Plot No. 11


BACK PRESSURE AS A FUNCTION
OF
POWER OUTPUT
Throttle position varied to change Engine Power. Exhaust passage conditions fixed for each curve but differing for the various curves.
End Outlet Valve Open, Aug. 24 & 29th.
New Exhaust Piping, with Muffler on.
New Exhaust Piping, Muffler off.
Actual Back Pressure, inches of mercury.
Brake Horsepower.
Plot No. 12



MANIFOLD & MUFFLER COMBINED

Upton-Gage idea. Adapted for H.S. Eng.

Fig. 13.

after taking the data given in Table II, the exhaust piping connections were changed somewhat in order to try out a muffler. The wire-gauze-filled muffler, described on page 48 and figure 4 of Report No. 10, was put on the open end of the exhaust manifold piping. Runs were made with and without the muffler as resistance, giving the data shown in Table III, and in plot 12. This is comparable with plot 7, the lower curve of plot 7 being reproduced as the upper curve of plot 12. All three curves are evidently of the same type. *Hence the information obtained from Tables I and II and their analysis will be applicable to the analysis of muffler actions so far as back pressures and power losses are concerned.*

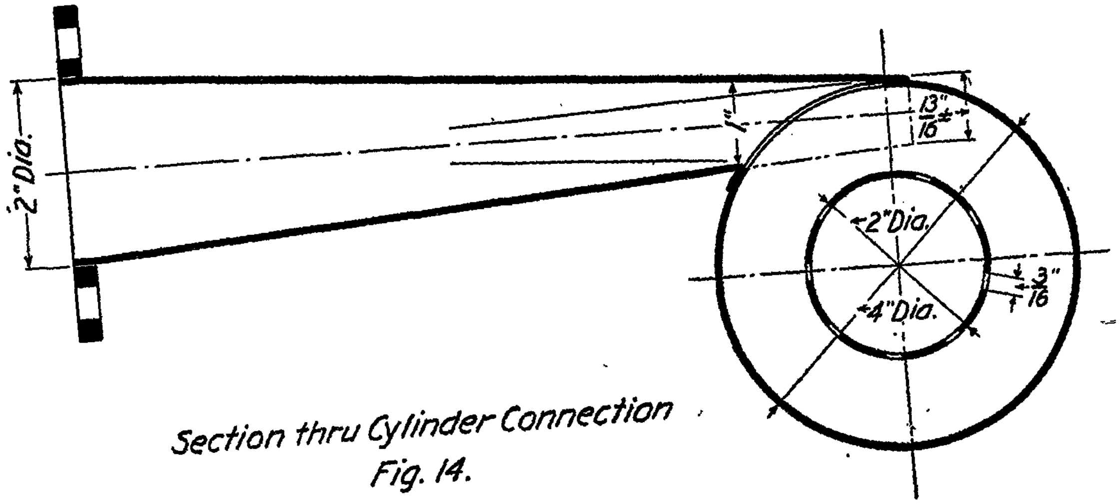
In the course of the experimental work so far described some peculiar phenomena were noted. One such was an abnormal power drop, considering the back pressure, at certain "critical speeds." It was found that this abnormal power loss was avoided by a very small change of speed either way from the critical. The critical speed changed or disappeared with change of exhaust manifold. Apparently some manifolds would not show this phenomena; probably the curved manifolds would be free from it. The supposed cause of this abnormal power loss at a critical speed is a reflected wave of exhaust gas filling the clearance of some cylinder just before its exhaust valve closes.

Study of this effect led to a suggested design of an exhaust manifold, and ultimately to the design of a muffler. If the pulses of exhaust gas, following the opening of successive exhaust valves, could be so trapped that they would never return to their own or other cylinders by direct motion, or by reflection, interferences and abnormal power losses would be avoided, and the muffling problem itself might be simplified. The scheme selected was rather similar to the muffler design shown in figure 3, page 47, of Report No. 10. The exhaust of each cylinder was to enter, tangentially, the annular space between two concentric cylinders. In this space the exhaust gas, entering with high velocity and pressure, could spin around, dissipating its energy both by friction and by progressive leakage through numerous small holes in the inner cylinders. The inner cylinder, continued outward from the manifold, was to become the exhaust pipe. A design for a manifold-muffler of this type is shown in figure 13. The tangential entrance effects the trapping of the exhaust pulse, preventing its direct or indirect return to any cylinder. This is shown more in detail in figure 14.

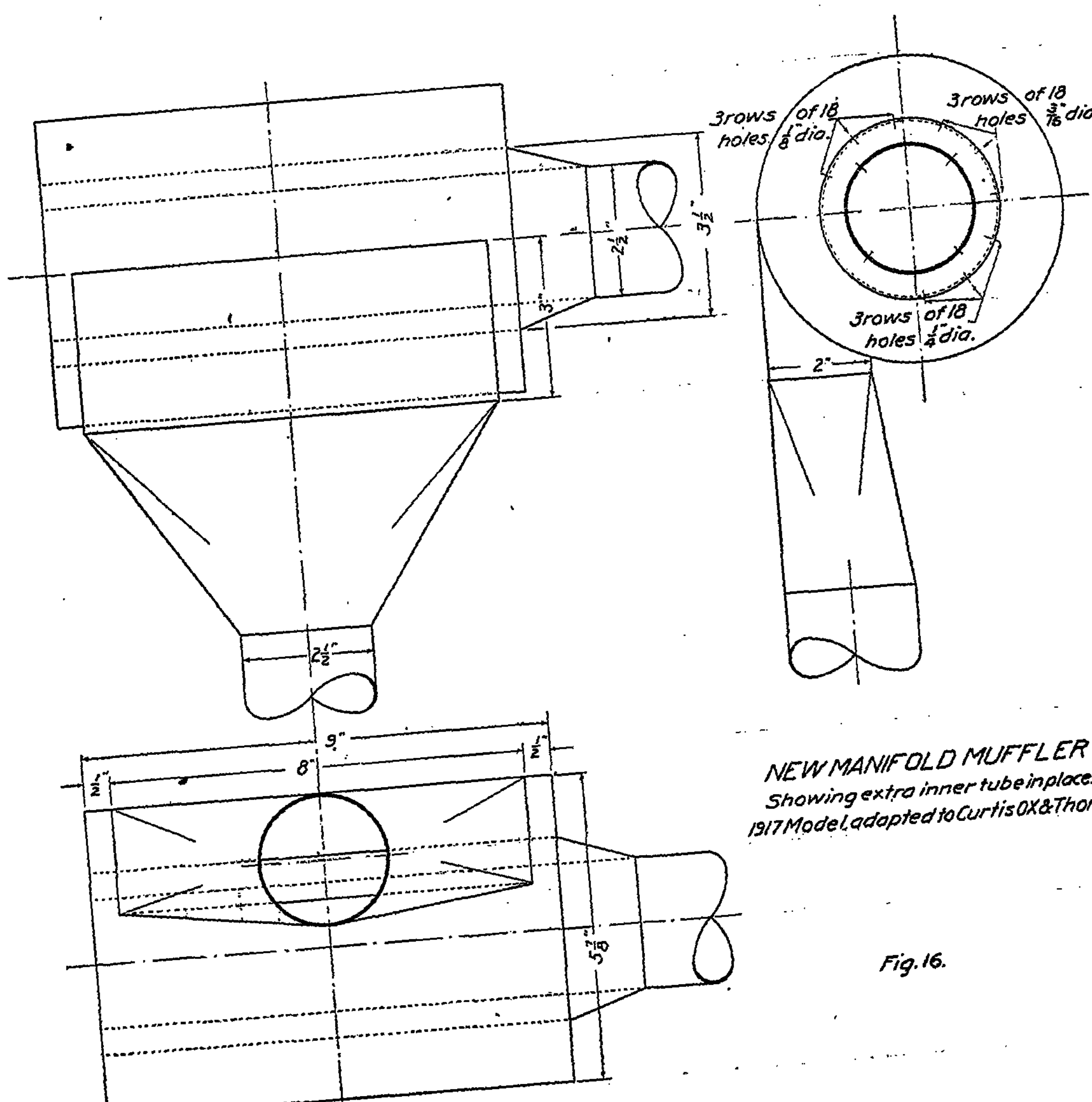
While a device of this character (figure 13) might be successful in normal operation of an automobile engine, it was discarded as unsuited for airplane engine service. First, the internal construction would burn out owing to the great heat of the exhaust gases in airplane service, and secondly, the radiant heat from the large surface of the manifold would prohibit the installation. The first objection may be met, and the second minimized, by the more compact construction suggested in figure 15, which could well be built with a smaller external diameter than the other design.

The remainder of the experimental work consisted of trials of various commercial mufflers and of experimental mufflers designed and built by the authors of this report. The experimental apparatus remained, in general, unchanged. The dynamometer, the tachometer, and the arrangements for reading back pressures remained the same and were used as before. Power losses due to the application of mufflers were found as described previously in the determination of power losses due to applied back pressures. When the emphasis was upon the determination of muffling ability, and the comparison of various mufflers with each other as silencers, the determination of power loss was inferred from the back pressure reading and general engine speed. This method is more accurate than the direct determination of power loss from speed drops, because the power losses from the mufflers finally selected were small. The drop of speed was too small to read satisfactorily while the back pressure was still a readily measurable quantity.

Usually, when working with mufflers, the Curtiss engine was set to run at its recommended speed of 1,200 to 1,250 r. p. m., developing about 70 horsepower. A scale of throttle settings had been made so that this, and other settings, could be duplicated. Generally one muffler was used for the entire engine, taking the exhaust from all eight cylinders. The setup shown in figure 1 was then modified by putting the quick-opening valve on the side outlet of the tee on

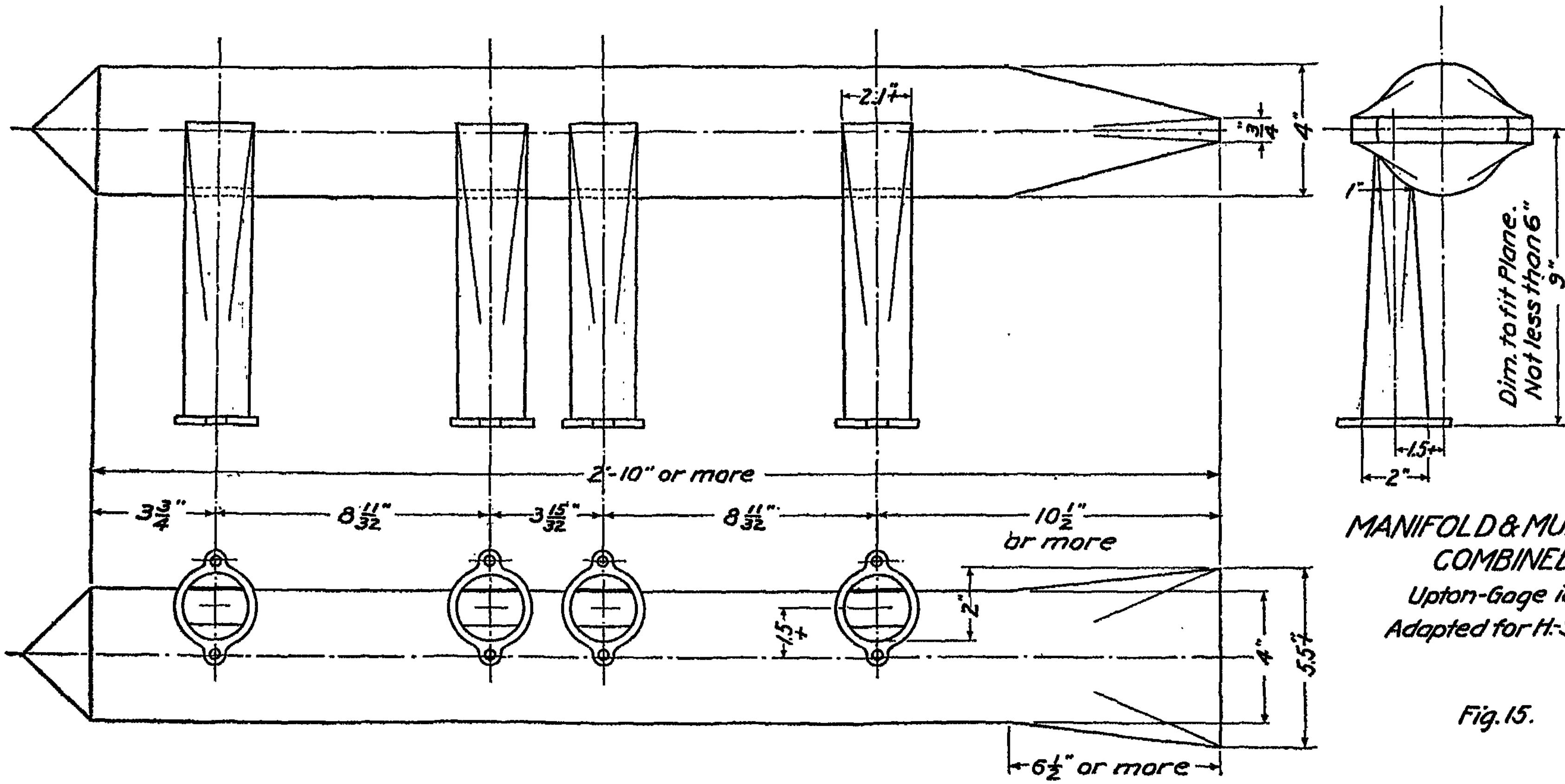


Section thru Cylinder Connection
Fig. 14.



NEW MANIFOLD MUFFLER
Showing extra inner tube in place.
1917 Model, adapted to Curtis OX & Thomas

Fig. 16.



MANIFOLD & MUFFLER COMBINED
Upton-Gage idea.
Adapted for H-S. Eng.

Fig. 15.

the end of the cross manifold, allowing the installation of the muffler under test, without a valve, in the direct line of the exhaust flow. The back pressure due to the muffler could then be determined, by taking alternate readings as before, with the quick-opening valve open and shut. The data of Tables III, VI, and part of VII came from this arrangement.

In a few tests, concerning particularly the capacity of mufflers, the cross manifold was removed. A muffler could then be placed directly on the end of the stock exhaust manifold for one block of four cylinders. Generally a muffler was placed on each of the two manifolds. The mufflers then had to be applied and removed by hand in order to determine the back pressure due to the muffler. This procedure was used for getting the data of Table IV, using the G. P. F. 12-inch and 28-inch mufflers, and part of Table VII.

Estimates were always made of the silencing qualities of the muffler being tested. In the end the devices selected as most promising were put through comparative tests, where rapid substitutions were made by hand. For this work it seemed desirable to make an open-air estimation of the noise of the exhaust and the degree of silencing with the various mufflers. To get the exhaust outdoors and pointed in the right direction the tee and valves on the end of the cross manifold were removed. At the end of the cross manifold direct connection was made to a 2-inch piping system consisting of a 6-foot length of pipe, a 45° ell, and a second 6-foot length of pipe. This piping did not muffle the exhaust to any noticeable degree. The back pressure due to the piping, at normal engine speed, was about 1 inch of mercury. This measurement, in itself, may be of considerable interest in the problem of the disposal of engine exhaust on airplanes. The data of Tables VIII and IX were taken with the mufflers placed outdoors in this manner.

Many devices and ideas for silencing were considered, and quite a few were tried out. All devices with which tests were made, and which were found worthy, as well as a few which were rejected, are included in the descriptive matter which follows. Of the rejected schemes, only those are described which are based upon some peculiarly attractive idea, or which are akin to common practice. The experimenters (the authors of this report) applied quite peculiar descriptive nomenclature to some of these devices. For the sake of brevity, some system of naming is necessary, so those designations will be perpetuated, and the derivation of the appellation briefly indicated.

G. P.-F.—These mufflers are shown in figure 2 and described on page 43 of Report No. 10. Four of these were used, two each of the two sizes. All were 5 inches diameter, two were 12 inches long, the other two were 28 inches long. They were made by Geuder, Paeschke & Frey Co. These are regular stock mufflers.

Maxim.—A Maxim silencer for Fords was purchased of a local garage. The entrance to this muffler was only 1.5 inches internal diameter. The tail pipe was 12 inches long, tapering from 1.5 inches diameter at the muffler to 1 inch diameter at the outlet. It was hardly fair to use this on a 70 horsepower engine; but the data is of great interest.

Manifold mufflers.—The name comes from the original idea of putting several of these devices in line, end to end, one for each cylinder, the combination to replace the exhaust manifold. However, this manifolding scheme was never thoroughly tried out; the one unit of the device was used as a muffler, with conventional installation at the end of the manifold. As originally constructed the inner cylinder did not touch the end plate, so that the exhaust could escape around the end as well as through the holes of the inner cylinder. The end path could be blocked by a piece of asbestos board fastened to the end plate. This first manifold muffler was of the same general design as shown in figure 16 and figure 17, but with the outer cylinder 6 inches long instead of 9 inches, and containing only the $3\frac{1}{2}$ inch perforated inner tube.

The "Long Manifold Muffler" was made up of two outer sections of the first manifold muffler, making an outer cylinder 12 inches long. The inner tube was not perforated and extended the full length of the muffler, its serrated end touching the far end plate. The exhaust entered the outer cylinder near one end through the regular tangential entrance of this type of muffler. The entrance of the second unit was blocked. The annular space was both spin and expansion chamber for the exhaust gases, which escaped into the inner tube only through its serrated end, at the opposite end of the muffler from the gas entrance and exit.

Later a "New Manifold Muffler" was built similar to the sections which were first made, but 3 inches longer, and with provision for slipping an extra perforated inner tube of 2½ inches diameter inside of the regular 3½-inch inner cylinder. This last design of the manifold muffler is shown in figure 16, with the extra inner tube in place, and in figure 17 assembled and disassembled. The manifold mufflers are developments of the muffler shown in figure 3, page 47, of Report No. 10.

Wire mesh type.—Several of this type were built and tried out. The general scheme is shown in figure 4, page 48, of the Report No. 10. The idea is to gradually increase the resistance, and to break up the energy, of the exhaust by means of sections filled with fine wire mesh. The finest mesh was used at the outlet, and the mesh increasing in size toward the engine.

Spiral guide vane.—This scheme consisted of an expansion chamber, with spirals in the annular outlet passage, instead of the wire mesh. The spirals formed, in effect, a long nozzle. Various forms of spiral were tried: of constant pitch, or area of passage; with area of passage growing smaller toward the outlet, forming a sort of converging nozzle, and with the area of passage increasing toward the discharge end. The general design of these is based on that of the Maxim silencer, but there is not the repeated reversal of flow in the spirals.

Whirl chamber mufflers.—So called from their general construction. Figures 17, 18, 19, and 20 show the form of these mufflers. The exhaust pipe is flattened from a circular cross section at the engine to a rectangle at the muffler, giving a contracted nozzle effect. This rectangular section is fastened tangentially to the circumference of a ring of considerably larger diameter. One cover plate for one side of the ring is a plate dished to give stiffness. The other cover is the outlet for the gas, and consists of a truncated cone. The exhaust gases enter the ring tangentially, swirling around and around inside the ring. As they lose velocity they gradually escape through the opening between the dished cover plate and the end of the truncated cone. Four of these mufflers were made, one each with rings 4 inches and 12 inches in diameter, two with 7-inch diameter rings. The 4-inch and 12-inch diameter were made with the idea of having one of them too small and the other too large. The exhaust pipe entering the muffler was contracted from a 2-inch diameter to a rectangle 1-inch wide, except that one 7-inch ring had a nozzle ½ inch wide. The sides of the rectangle had the same total perimeter as the circumference of a 2-inch circle. This is a constructional requirement. The width, or depth, of the cylindrical ring was limited to the length of the rectangular discharge nozzle, plus clearance necessary in manufacture. This makes the rings 4 inches deep with the 1-inch nozzles, and 4½ inches with the ½-inch nozzle used on one of the 7-inch whirl chambers. The cover plates were dished about ¼ inch, this being the maximum obtainable with the local tinsmith. The design called for ½ inch. These plates could be applied with the convex surface in or out as indicated in figure 19. The area of the open end of the truncated cone was the same as that of the exhaust pipe, that is, 2 inches diameter. The space between the end of the truncated cone and the cover plate was varied from time to time by the various arrangements, some of which are indicated in figure 19. Shallow cones were available as well as the deep ones. With deep cone, and cover plate dished inward, the clearance through which the exhaust had to escape was about ¼ inch. With cover plate dished outward the clearance became ¾ inch. With shallow cone the clearance was 1½ inches.

The 7-inch whirl chamber muffler was also tried out with a double cone assembly, one cone pointed inward and one outward. (Diagram of arrangement in Table VII.) This assembly was also tried with a "diffuser" plate between the two cones. One diffuser was a plate with about 100 scattered $\frac{3}{16}$-inch holes punched through it. Another diffuser tried with both 7-inch and 12-inch whirl chambers had no holes, or very small ones, in the 120° sector first passed by the entering gas, the next 120° with larger holes, and still larger holes in last 120°.

The *Duplex whirl chamber muffler* consisted of the two 7-inch rings bolted end to end, using one of the tangential nozzles as entrance, and the other as discharge passage. The arrangement required the reversal of internal whirling before the gas could get out. Generally one or the

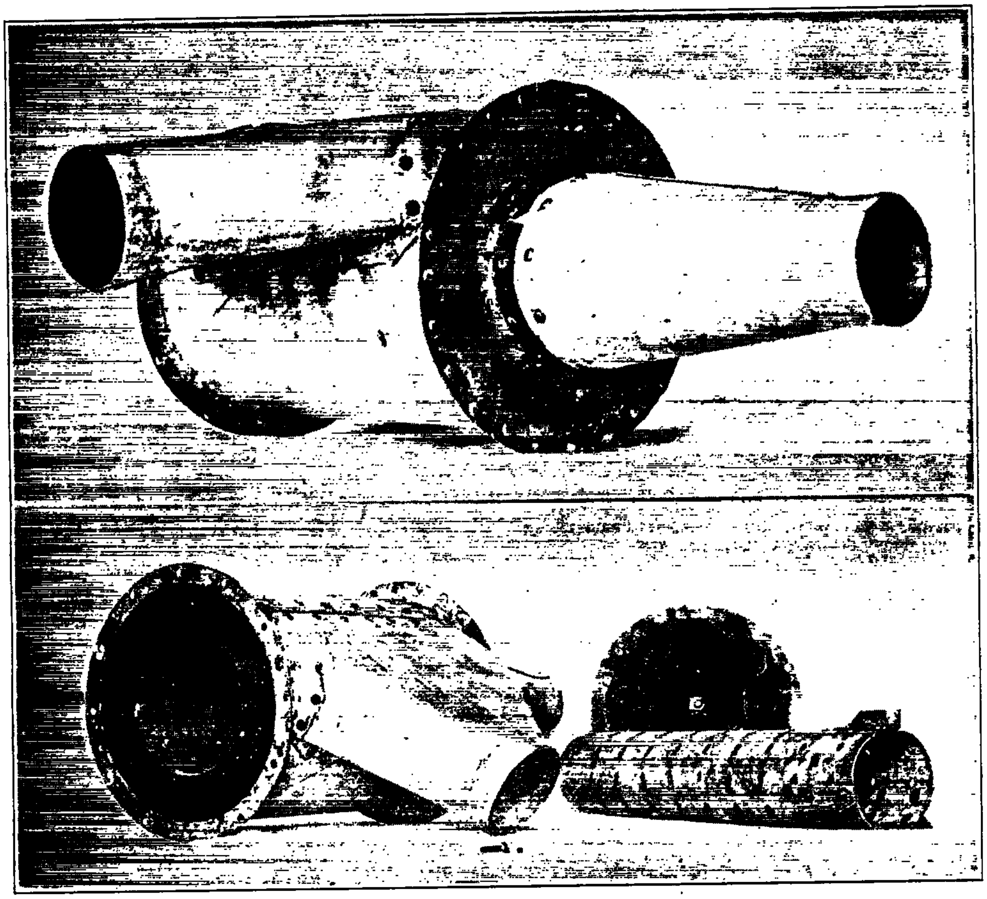


FIG. 17.

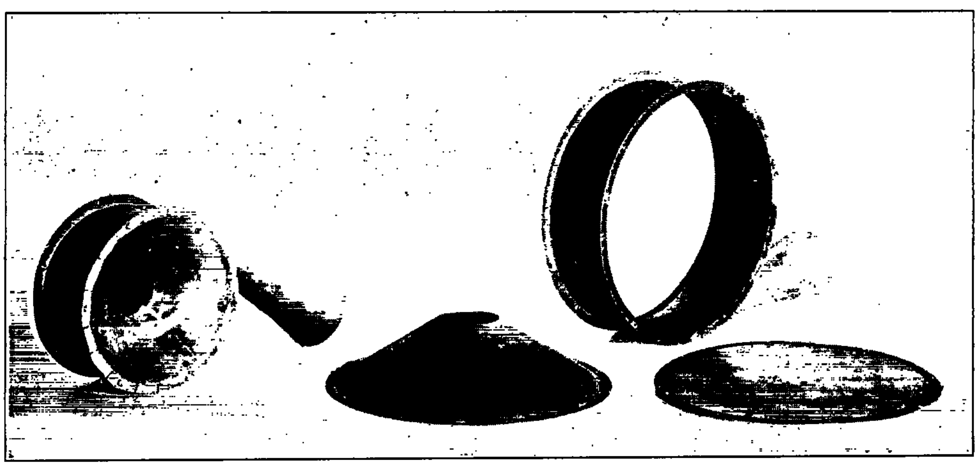
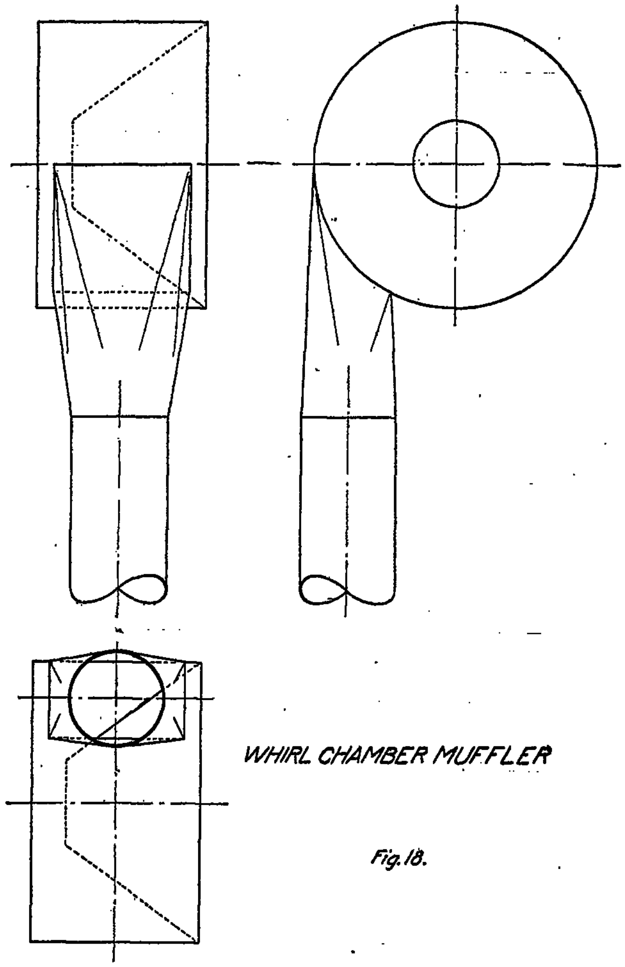


FIG. 20.

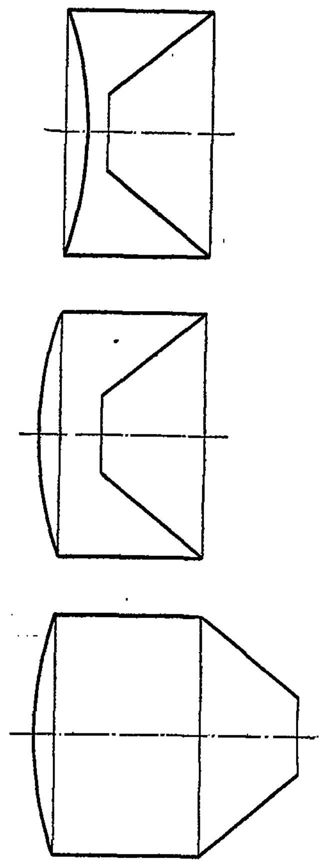
other of the diffuser plates was placed between the two rings. The 1-inch nozzle was used as entrance in most cases.

Venturi.—A 2-inch pipe Venturi with ¾ inch throat was tried, upon the idea of an expanding nozzle to secure an adiabatic drop of pressure and temperature of the gas, also, to increase the velocity of the gas above the velocity of sound, so as to prevent any sound waves issuing from inside the manifold. The Venturi was used alone, and in combination with the 7-inch whirl chamber muffler, the Venturi then acting as a discharge pipe.



WHIRL CHAMBER MUFFLER

Fig. 18.



VARIOUS ASSEMBLIES OF WHIRL CHAMBER MUFFLERS

Fig. 19

The problem of silencing the exhaust noise from an internal combustion engine is fairly comparable with the problem of silencing a high velocity rifle, or better, a machine gun. At the instant that an exhaust valve begins to open, the gases seeking to escape from the cylinder have a pressure in the neighborhood of 40 pounds per square inch gauge and temperatures of the order of 1,000° F. Where the endeavor is to get maximum power from the engine, these values are understated. If adiabatic expansion is assumed at the time of release, the initial velocity of the discharging gas is independent of the pressure into which the gas is escaping, because the "critical pressure" is greater than the pressure in the manifold or atmosphere. The initial velocity will be the velocity of sound at the "critical" pressure and existing temperature of the gas at this pressure. This velocity of the exhaust gas is of the order of 1,500 to 2,000 feet per second—considerably higher than the velocity of sound through the air. The first portion of the escaping exhaust is practically a slug of gas coming out like a projectile from a gun, with a velocity greater than that of sound in air.

It is well recognized that the report of a high-velocity rifle consists of two sounds. One sound wave, the "report," comes from the muzzle of the gun arising from the slap of bullet and exhaust gas upon the air adjacent to the muzzle. The other sound, of cracking, ripping quality, comes from the tearing of the air by a projectile moving with a velocity in excess of the velocity of sound. If the bullet is traveling toward the observer, this second sound is heard first. The slugs of exhaust gas from the open exhausts of an airplane motor are soon dissipated, and do not travel far as projectiles. While they last, however, there is little doubt that they contribute to the quality and amount of exhaust noise from the engine.

If the velocity of the slugs of exhaust gas at the time when they enter the air can be reduced below the velocity of sound in air, the quality of the exhaust sound will be much duller and the quantity of noise much less. The most effective way of lowering the velocity of sound in a gas is to lower its temperature.

There are two means of accomplishing the cooling of the exhaust gases before they reach the atmosphere. One is by direct cooling, as by water jacketing. The other is by designing the exhaust passages on the idea of the nozzles of a single velocity stage steam turbine, so as to reduce the velocity of the exhaust gases to approach the velocity of sound in air, by efficient expansion in the nozzles. This latter method was suggested by Mr. F. C. Mock. (S. A. E. Bulletin, p. 270, Vol. V, No. 3, Dec., 1913.) It was found that such design of exhaust ports apparently increased engine power. However, the reduction of temperature by expanding in nozzles is limited, if no work is done by the gas. Only the edge of the crack of the noise from the slug of gas can be removed by means of expansion alone. If the exhaust is made to drive a turbine, then more heat energy will be abstracted, useful work will be done, and at the same time the exhaust velocities will be made uniform and small.

The frequency of the exhaust from a multicylinder, high-speed engine generally is a source of a humming sound, if in any way the air is made to vibrate with the engine frequency. The lowest audible note to the human ear is of about 40 beats per second. The exhaust frequency of a 12-cylinder 4-cycle engine running 1,500 r. p. m. is 150 per second. This will be a low-toned hum. To this hum an airplane propeller also contributes harmonics which will blend with the engine hum.

Complete muffling, so that the engine will give neither crack, whistle, nor hum of exhaust noises would involve the smoothing out of the flow of exhaust gases into the atmosphere to a uniform velocity below that of sound. This is manifestly impracticable. With a pulsating flow it might be practicable to keep the maximum velocity of exhaust gas entering the air below the velocity of sound in the air. The humming noise will then be heard, without the crack or whistle.

The greater portion of the exhaust from a cylinder must pass out in this first slug of high velocity discharge, as the average velocity of the piston on the exhaust stroke is only about 25 feet per second. This shows that the actual time required for discharging the major portion of the exhaust of any one cylinder is quite short compared to the period during which the exhaust valve of this cylinder is open. A whole group of cylinders may be discharging into one exhaust manifold without interferences, provided the manifold is so designed that each slug of exhaust can freely escape down the manifold without check or reflection. If too many cylinders exhaust into one manifold, there will be overlap of the scavenging periods, during which two or more of the various exhaust valves may be open at the same time. This overlap of the scavenging periods is a minor matter compared to what happens when a slug of exhaust enters some cylinder through an open exhaust valve directly or by reflection.

An exhaust manifold may reduce the crack of the exhaust by slowing down the initial velocity of the slugs of gas, and by a limited amount of cooling. The direction in which the exhaust is pointed is a considerable factor in the noise heard by observers, the intensity of sound being much greater in the direction of projection. Long exhaust pipes may somewhat increase the muffling effect of a manifold, by added friction and added cooling. However, the notation on Table VII, based upon the estimates of several observers, would indicate that muffling effect of manifold and long smooth exhaust pipe was very slight. Bends, and also rough interior surfaces, will increase the muffling effect and also the power loss.

Any long pipe may act as does an organ pipe, having a natural period of vibration in sound waves. This period may happen to coincide with the frequency of the exhaust in such a way that exhaust gas will be forced back into some cylinder just before the closing of its exhaust valve, entailing an abnormal power loss at a critical speed. Such an effect was noted in the experimental work, and was referred to in the discussion of Table II.

As previously stated, the silencing problem would be solved if it were feasible to reduce the velocity of exhaust gases escaping into the air to a uniform velocity, less than the velocity of sound. Practically, it is possible only to spread out the peak of the discharge, or the "slug" of exhaust pulse, so that its velocity is reduced toward that of sound. Every slug of exhaust, not only while in the exhaust system but in the period of issuing from the system, is a potential source and exciter of sound waves. The internal vibrations of the system as excited by each slug, combine their sound with that of the impact of the slug upon the air, and change the quality of the sound as a whole. Great irregularities of internal form of the exhaust system may break up the internal sound waves. The sound-producing qualities of the exhaust slug are relatively much greater than the sound waves internal in the exhaust system.

A very obvious device for dissipating the excessive velocity of initial flow of each exhaust is an expansion chamber. The gas velocity entering the chamber is highly irregular; it is assumed that the discharge velocity is much more nearly uniform; and the slapping of the external air by successive exhaust slugs is stopped. This is nearly true if the chamber is capacious enough. The slapping action takes place as theexhaust enters the expansion chamber, and sound waves from this point radiate outward, some of them escaping through the tail pipe. To complete this device as a muffler, the exit of sound waves would have to be prevented, it being assumed that the capacity of the chamber is sufficient to prevent the slug passing through as such.

If through the exhaust line there is placed a long series of baffles, each baffle will drag back a portion of a passing exhaust slug, delaying this portion with reference to the remainder, and thus changing the flow from intermittent to nearly uniform. Muffling might then be secured by a sufficient amount of baffling alone. The cost in back pressure and power loss would be prohibitive, to say nothing of the weight of material necessary for the baffles and general structure. When baffling structures are used, the highly heated exhaust gases burn out the internal structure, and carbon deposits and oxide scale choke the baffles. This burning away of internal parts is a very serious objection to the use of baffles in mufflers for airplane motors.

A combination of expansion chamber and baffles can be made into a very satisfactory muffler. The smoothing of the pulsations of the exhaust flow is done mainly by the expansion chamber, and with much less power loss than if it were done by baffles. The baffles permit the expansion chamber to be made of moderate dimensions, by helping the smoothing out of the flow. Also, if properly designed, they will nearly prevent the escape of sound waves from the expansion chamber by reflection (backward) and dispersion (scattering and interference) of the sound waves coming from the initial slap of the slug.

In order to get the effect of a large expansion chamber without great weight and size, the manifold muffler type ("whirl chamber") construction was proposed. The exhaust is brought tangentially into an annular space between two concentric cylinders. The slugs of exhaust gas may continue to travel around in this space, but they can not escape from this cylinder until their velocity is sufficiently reduced to reverse or change direction in some way. If, for example, the gas slug whirls around the chamber 20 times (it was observed to travel around more times than this upon a single cylinder slow-speed engine) it has had, in effect, the use of an expansion chamber of a volume 20 times the volume of the annulus. As the gas spins, losing velocity, a continued series of small portions escape, at right angles, into the inner cylinder or discharge passage. The gas can more readily turn at right angles than reverse itself, so that there is little chance for a return pulse to the cylinder. In spinning around and escaping inwards, a subdivision of the exhaust pulse is made such that the successive small portions enter the atmosphere over a comparatively long period of time, making the velocity of the exhaust gas entering the air fairly uniform instead of highly intermittent. It is also probable that the initial slap of exhaust entering the whirl chamber of the mufflers is less than if they abruptly entered a large expansion chamber, because there is relatively a much smaller change of size of passage. As contrasted with the use of baffles to smooth out the exhaust pulsations, the desired end is accomplished in these "whirl chamber" designs (and "manifold mufflers") with relatively less power loss through back pressure.

In the typical "whirl chamber mufflers" all metal parts had one side exposed to the air in order to have considerable cooling effect upon the exhaust, and also to avoid the danger of the burning away of the metal. (The success of this cooling is illustrated upon one of the mufflers tested, which still has upon it a paper label only slightly charred.) In so far as any cooling of exhaust is secured, the muffling is thereby aided.

A sufficient cooling of the gas will, in itself, silence the exhaust. Such cooling is only possible in marine practice, where water is sprayed into the exhaust line, and is found effective. It may be desirable to place fins for air cooling upon the exhaust manifolds, pipes, and mufflers of airplanes. A radiator used to cool the exhaust gases for the purposes of silencing may sound like a humorous suggestion, and yet it may be practical on heavy duty planes.

Before a muffler can be applied to an engine there must be a manifold of some kind to take the exhaust gases from the engine to the muffler. This manifold is itself a source of power loss, both through friction of flow, and through the possible interferences already mentioned, particularly the back flow of exhausts. In the early days of internal combustion engines, manifolds were first made by bringing the pipes from the individual cylinders squarely into a collecting pipe. The right angled turns of this design caused high back pressure and promoted interferences. To decrease the back pressures (and interferences) the manifolds are now made with sweeping curves on the discharge pipes of each cylinder, making each individual pipe have an easy entrance, in the direction of flow, into the common pipe. It is suggested that a more compact design could be made by analogy with the "whirl chamber mufflers." The individual exhaust pipes might be brought straight out from the cylinders, but with their center lines so far above or below the center line of the common pipe that the exhausts would make tangential entrance to the common pipe. This idea is shown in the designs of figures 13, 14, and 15. The change of direction of flow from the individual pipes to the common pipe can probably be made with back pressures no larger than from the sweeping bends of the conventional construction, and with the advantage of compactness. The smallness of back pressures may be inferred from the data at the bottom of Table VI on the resistances of rings with nozzles only.

Throughout all the design of manifolds and mufflers there is one item that must be kept continually in mind. The sharp pulse of each exhaust is practically a mechanical slap or blow upon all of the inside surfaces of the metal parts. If these metal parts are made of thin material, as they must be to save weight, it is necessary to so form them that they are inherently stiff, incapable of buckling or drumming. Otherwise they will become transmitters for the exhaust sound to the adjacent air, with additional noises from the reverberation of the metal itself. Flat surfaces are to be avoided, and doubly or singly curved surfaces chosen.

The preceding paragraphs have outlined the theory of muffling and associated problems as it developed to the authors during and after their experimental work.

Early in the work there arose the question of how the capacity rating of a muffler should be made. The data of Table IV precipitated this question. The two different sized mufflers compared were presumably alike in internal design save that the longer ones contained a greater number of the baffling elements in series. The G. P. F. design is given in figure 2 of Report No. 10. According to the makers the horsepower capacity to be handled by the 28-inch mufflers is four times that of the 12 inch. They recommended the 12-inch mufflers for engines up to 553 cubic inches displacement, and the 28-inch mufflers for engines up to 138 cubic inches. Yet it was found that on the Curtiss engine, with one muffler handling 4 cylinders, of 251 cubic inches displacement, the 12-inch mufflers gave 2 inches of mercury back pressure, against 4 inches from the 28-inch mufflers, and if there was any choice as to silencing ability, the smaller mufflers were the better. It was also found that the 12-inch G. P. F. muffler failed to silence a $3\frac{11}{16}$ by 4 inch Chevrolet automobile motor of about 170 cubic inches displacement. This same 12-inch G. P. F. muffler gave the same back pressure and better silencing effect when handling all 8 cylinders of the Curtiss engine (Table VIII) 502 cubic inches displacement, that it did when handling only 4 cylinders of the same engine.

In Table VII the data shows that one of the "manifold mufflers" had this same peculiarity of improving its silencing, and of not increasing its back pressure when the number of cylinders handled was changed from 4 to 8. At the same time some "whirl chamber mufflers" increased their back pressure in the ratio of 1 to 1½ or 1 to 2 when changed from handling 4 to handling 8 cylinders. If the back pressure of a muffler followed the usual laws controlling the increase of pressure with quantity passing, the doubling of number of cylinders exhausting into a muffler should have multiplied the back pressures by 4. The explanation is in part in the design of the mufflers themselves, and in part due to the peculiarly intermittent flow of the exhaust gases.

The silencing by the muffler comes in its operation upon the "slugs" of the exhaust gas, the size and character of which is fixed by the individual cylinders. In so far as the frequency of impulse (due either to multiplication of the number of cylinders or to the increase of speed) is concerned, it appears that the more frequent the impulse, the easier it is to silence the exhaust noise. The power loss, or back pressure, seems to depend upon the form and size of the individual impulses. Perhaps if the number of cylinders should exceed the present limit (12) there might be sufficient overlap of instantaneous impulses to require a larger muffler. The bulk of the exhaust comes at the first opening of the valve, so these pulses are not liable to be superposed to any appreciable extent. The characteristics of the individual exhaust pulses are controlled by volume of cylinder, valve timing, throttle position, ignition timing, and mixture ratio, all of these affecting the amount of gas and its pressure at the time of opening the exhaust valve. The speed of an airplane engine is tied with the throttle position because of propeller characteristics, while an automobile engine throttle and speed are independent.

As a matter of fact, it appears that the smaller the mufflers, up to some limit, the better the silencing. Also the larger the muffler, the less the back pressure, geometric similarity being assumed. (The 28-inch G. P. F. muffler is not similar to the 12 inch.) The effect of change in back pressure is slight. So that the tentative conclusion is reached that, with the muffler design geometrically fixed, and if the size only is changed, then the smaller the muffler the better the silencing and all around action, until the power loss exceeds the tolerated value.

It may be remarked that mufflers taking all 8 cylinders of the Curtiss engine receive about the same frequency and magnitude of impulse as if used on one side, 4 cylinders, of back geared motors such as the Thomas or Sturtevant.

The effect of size of muffler was noted when using the same mufflers on different engines. The greater the bark of the exhaust, the better was the relative suppression by the same muffler. The mufflers were more effective in suppressing the bark of the single cylinder farm engine (5½ by 10 inch Ingeco) than on the Curtiss (4 by 5 inches).

The suggestion from the facts mentioned above is that the capacity rating of a muffler probably should not be based upon the total displacement of the engine, so much as on the displacement per cylinder.

In reading the discussion which follows concerning the various schemes for silencing the exhaust which were considered in this work, it should be kept in mind that the experimenters had formulated certain requirements and limitations for mufflers in airplane service. The manifolding and mufflers should not be a source of fire risk from radiant heat. Muffler explosions should be made harmless, either through sufficient strength, or provision of a breaking piece. Weight of manifolds and mufflers must not be excessive. Any parts so disposed as to cause head resistance must be made as small as possible and of "stream line" form. The power lost due to back pressure must also be very small. It is desirable, but not essential, that the muffler be durable, especially with regard to burning out the interior parts. The amount of silencing required is not great, compared to the usual ideas of muffling devices as exemplified on automobiles. The reason for not requiring so effective silencing is that there are many other noises coming from a plane moving through the air. Such noises are the hiss or whistle of the wires, the beat or drum of the propeller, and the valve and gear noises from the engine. It has been assumed in this report that what the experimenters and assistants called a 50 per cent total silencing of the exhaust noise would be sufficient and satisfactory for airplanes. If there is more silencing than this, a muffler cut-out may be needed by the pilot to judge the engine performance. However, it was found that 75 per cent silencing was easily obtained, so this was soon adopted as a standard.

A common device on planes is a long exhaust pointed upward. This is not a muffler, but it is somewhat effective in directing the sound away from the ground. Long pipes running back along the fuselage take the exhaust gas and some of the noise away from the occupants of the plane. If pipes of smooth interior are used, the power loss due to back pressure is relatively small. In Table VIII it is noted that 12 feet of 2-inch pipe with one 45° elbow caused a back pressure of 1 inch of mercury on a Curtiss (8-cylinder OX) engine, resulting in a power loss of about seven-tenths of 1 per cent. It is regretted that data on the back pressure due to flexible metallic exhaust hose was lost, but the notation made was that 4 feet of the 2-inch flexible hose gave more back pressure than the 12 feet of standard 2-inch wrought-iron pipe including one 45° elbow.

The first muffler tried out in this series of tests was a wire mesh muffler shown in figure 4 of Report No. 10. The muffler type is that of expansion chamber plus baffles, relying largely upon the baffles for noise suppression. The power loss was slight and the muffling estimated about 50 per cent. The weight is 15 pounds, which is comparatively great. The flat metal sides probably drummed, reducing the muffling ability. It was anticipated that the wire gauze would burn out and choke the passages with scale under continuous operation. Even in short operation the wire gauze began to pack although it did not burn. These prospective troubles, together with the weight, caused the discarding of this type.

The variable pitch spiral muffler mentioned in Table V, was, as constructed, cumbersome. Its silencing was estimated at 50 per cent plus. The construction embodied a cylindrical expansion chamber plus a baffle placed in a concentric annular space. The baffle was made of a single strip, helically wound around the chamber, making, in effect, a long unobstructed path. This device is of the same type as a number of commercial mufflers which have already been more highly developed. It is subject to the disadvantages of weight, size, and burning out.

A Venturi with throat 0.75 inch diameter and of 2-inch entrance and exit was tried on the end of the 2-inch exhaust pipe. The throat size was selected to give a gas velocity greater than the velocity of sound, in order to prevent the transmission of sound from the engine through the exhaust pipe. The success was undoubted, as there was no crack. But the velocity of the gas in the throat set up an unearthly noise all its own. The expanding portion of the Venturi, instead of serving as a diverging nozzle, acted as a megaphone. Then the Venturi was tried with the exhaust first passing through the 7-inch "whirl chamber" muffler, with somewhat similar results. The Venturi was also rejected.

The Ford Maxim muffler was found to be a very effective silencer when used upon the Curtiss engine, as well as when used on a Ford automobile. On the Curtiss engine, handling all 8 cylinders, the power loss was rather high compared with some other devices. Considering the fact that the muffler was built for a small engine (the tail pipe was 1 inch diameter), this is hardly to be wondered at. The weight of 12 pounds was prohibitive.

Throughout the experimental work the G. P. F. 12-inch muffler was used as an arbitrary standard of muffling qualities. It appears repeatedly in the tables on account of this. While estimates of noise suppression were attempted in absolute values, the final decision always rested upon relative performance. The "Remarks" of Tables VIII and IX will illustrate this. The G. P. F. 12-inch muffler construction, as shown in figure 2 of Report No. 10, is novel. The baffles, in the form of nozzles, occupy the expansion chamber. The parts are few and simple. Surfaces are doubly curved, making for inherent stiffness. The peculiar fact was noted that this muffler worked equally well with either end as entrance, both as to back pressure and silencing. There is a possibility that the internal parts may act as dispersers of the sound waves by reflection, as well as other ways.

Our attempts to design mufflers, especially adapted to airplane use, have followed two main lines. The "manifold" series of designs use tangential entrance to an annular whirl chamber, the gases gradually escaping to the central part as they spin. The spin chamber gives the effect of the conventional expansion chamber with a much smaller volume. The perforated inner tube replaces the baffles, with less weight and back pressure. The objection to the construction is that the inner cylinder may burn out. To avoid this objection the "whirl chamber" series was designed. The action is essentially the same as the "manifold" type, but all metal parts are air cooled on one side, and the weight is less. A particular study of the "whirl chamber" type is given in Tables VI and VII. While making the observations on bark suppression, using the Ingeco engine (Table VI), the tests were made with load, at about 100 explosions a minute, and with retarded spark to give a vicious bark. The exhaust was flaming as it issued from the engine, and gave visual demonstration of the spinning action in the whirl chamber, and of the gradual emission of gas from the muffler.

Both "manifold" and "whirl chamber" types have a peculiarity that may be advantageous. The exhaust is turned at right angles between the entrance and discharge of the muffler, and with very little back pressure. Roughly, the back pressure is less than when using an ordinary pipe-fitting elbow. Short tail pipes of diameter equal to the manifolds were tried upon these types of mufflers, and were found not to affect either silencing or back pressure. So these mufflers might be placed at the end of a horizontal manifold, and the muffler tail pipe be carried vertically upward.

The data from the different tables in this report should not be indiscriminately compared. The manifolding conditions were frequently changed. In Tables VIII and IX the manifolding arrangements were the same, apparently, but actually were not constant. The increasing vibration of the engine continually shook loose the packing of the joints of the complicated exhaust line. Leaking of the exhaust from the manifolds became noticeable and evidently serious. Part of the piping was made of flexible metallic hose, from which the asbestos packing departed. These troubles make it improper to compare the data of Tables VIII and IX directly with the other tables, especially in regard to back pressure. However, the small group of tests, marked off by themselves in these tables, are correct for relative internal comparisons. To show how these increasing leaks affected back pressure results, the diagrammatic Table X is given. With the aid of this diagram, applied to the data given in the previous tables, the summing of results as given in Table XI is derived. Power losses are here inferred from the curve established in plot 2.

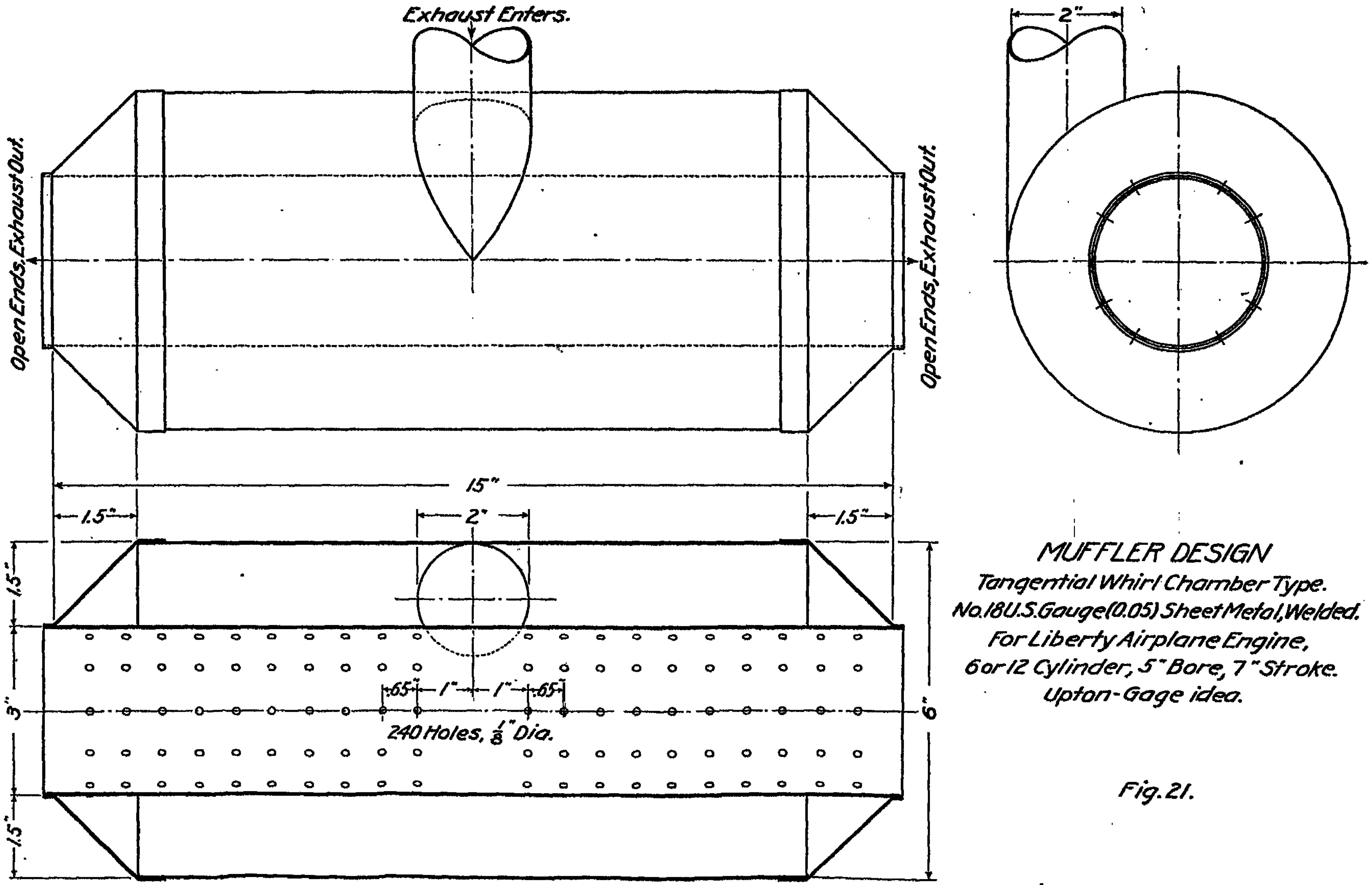
It is evident from this table that the "manifold" type of muffler will give good silencing with power losses less than 1 per cent and with weights comparing very favorably with any commercial muffler. If minimum weights are desired, the "whirl chamber" type looks most promising although its silencing action is not as good as the manifold type.

In the "manifold" muffler type the size, shape, location, and total area of the holes in the inner tube may be varied over a considerable extent. We used a total area of all holes equal about one-half the area of the exhaust pipe with very good results, and deviations did not alter the action of the muffler to any great extent, although many small holes probably gave better muffling without corresponding increase of back pressure. The best construction happened to have no holes opposite the entering gas, 33 holes ⅛-inch diameter which the gas first passed, 33 holes $\frac{3}{16}$-inch diameter next, and 33 holes ¼-inch diameter last, just under the nozzle. We do not lay particular stress upon the size or location, except they shall not be too big.

Taking all the results as shown in plot 10 into consideration, we have recommended, as a tentative design for the Liberty 12-cylinder engine, the design shown in figure 21. A variant of this is given in figure 22. Both designs may possibly be improved by the addition of cooling fins. They are supposed to be placed at the end of an exhaust manifold handling the exhaust from either six or twelve cylinders. The length of the muffler is to be parallel to the fuselage. There is no real attempt to stream line the back end of these mufflers for the stream of escaping exhaust gas is supposed to perform this function. Also any suction at the back end of the mufflers due to lack of stream lining may be worth its cost. In figure 21 is shown a design in which air is supposed to pass through the inner tube to some extent, aiding in cooling.

It is possible that very effective silencing could be obtained if the exhaust manifold were patterned after the design shown in figure 15, in combination with a muffler.

The completion of the solution of the muffling problem can only be accomplished by trials at fitting manifolds, mufflers, and tail pipes to engines installed in airplanes and in use in actual flight.



MUFFLER DESIGN
Tangential Whirl Chamber Type.
No. 18 U.S. Gauge (0.05) Sheet Metal, Welded.
For Liberty Airplane Engine,
6 or 12 Cylinder, 5" Bore, 7" Stroke.
Upton-Gage idea.

Fig. 21.

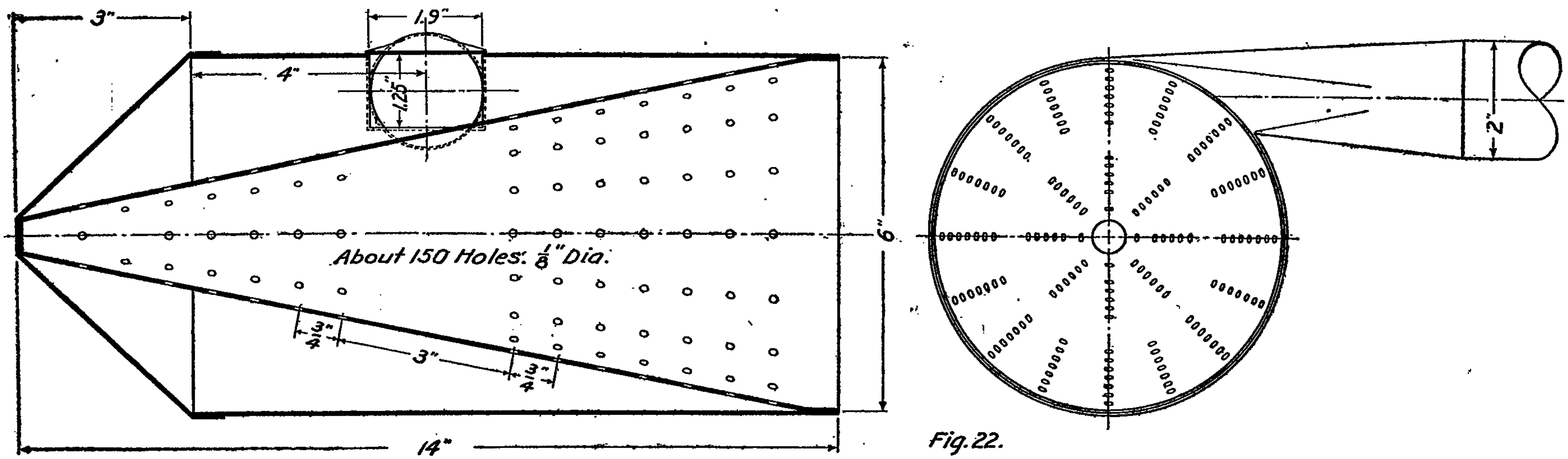


Fig. 22.

TENTATIVE SUGGESTION FOR MUFFLER

No. 18 U.S. Gauge (0.05) Sheet Metal, Welded.

For Airplane Engine, 6 or 12 Cylinder; 5" Bore; 7" Stroke.

TABLE I.—*Runs at constant throttle, varying back pressures.*

Throttle set to run the engine approximately at the power recommended by the maker, 70 B. H. P. at 1,250 r. p. m., with no applied back pressure except that due to the manifolds and piping. Table shows changes by applied back pressure. Data of Aug. 24 and 29.

Observations.				Results. (Based on average of observations.)											
Position of end outlet valve.	Back pressure (inches of mercury). South side.	Back pressure (inches of mercury). North side.	Tachometer reading (r.p.m.)	Position of end outlet valve.	Average tachometer (r.p.m.)	Average back pressure (in. mercury).	Applied back pressure (in. mercury).	Speed change, not corrected (r.p.m.)	Percent speed change (max. r. p. m. =100 per ct.)	Percent power loss.	True speed (r.p.m.)	Brake horsepower.	Brake M. E. P. (pounds per square inch).	Loss of brake M. E. P. due to applied back pressure.	
Shut........ Open........ Shut........	0.75 .50 .85	0.75 .45 .90	1,120 1,120 1,120	Open.. Shut..	1,120 1,120	0.47 .81	0.34	0	0	0	1,223 1,223	64.7 64.7	83.5 83.5	0	Aug. 24, 1916
Shut........ Open........ Shut........ Open........	1.6 .5 1.8 .5	1.7 .5 1.7 .5	1,130 1,135 1,125+ 1,130+	Open.. Shut..	1,135 1,130	.5 1.7	1.2	5	.44	1.3	1,239 1,233	67.3 66.3	85.7 84.9	0.8	
Shut........ Open........ Shut........ Open........	3.2 .5 3.4 .5	3.1 .5 3.3 .5	1,125 1,135+ 1,125+ 1,135+	Open.. Shut..	1,137 1,126	.5 3.25	2.75	11	.968	2.9	1,241 1,229	67.6 65.7	85.9 84.3	1.6	
Shut........ Open........ Shut........ Open........ Shut........ Open........	5.4 .5 5.8 .5 5.8 .5	5.1 .4 5.7 .4 5.6 .4	1,085+ 1,100+ 1,080+ 1,095+ 1,085+ 1,100+	Open.. Shut..	1,100 1,085	.45 5.57	5.12	15	1.364	4.0	1,201 1,184	61.2 58.7	80.5 78.2	2.3	
Shut........ Open........ Shut........ Open........ Shut........ Open........	8.2 .5 8.7 .5 8.6 .5	7.8 .4 8.3 .4 8.3 .4	1,085 1,100 1,075 1,105+ 1,075 1,100	Open.. Shut..	1,102 1,078	.45 8.31	7.86	24	2.18	6.4	1,203 1,177	61.6 57.6	80.8 77.3	3.5	
Shut........ Open........ Shut........ Open........ Shut........ Open........	10.6 .5 10.8 .5 10.9 .5	10.2 .4 10.5 .4 10.6 .4	1,030+ 1,095+ 1,025+ 1,100+ 1,030+ 1,105	Open.. Shut..	1,102 1,030	.45 10.60	10.15	72	6.53	18.3	1,203 1,124	61.6 50.3	80.8 70.5	10.3	
Shut........ Open........ Shut........ Open........ Shut........ Open........	10.2 .6 10.8 .6 10.8 .65	10.0 .5 10.4 .5 10.5 .5	1,070 1,115 1,065 1,120 1,065 1,120	Open.. Shut..	1,118 1,067	.55 10.45	9.90	51	4.56	13.1	1,220 1,164	64.2 56.8	83.1 75.6	7.5	
Shut........ Open........ Shut........ Open........	10.9 .6 11.0 .6	10.5 .5 10.7 .5	1,050 1,120 1,050+ 1,120	Open.. Shut..	1,120 1,050	.55 10.77	10.22	69	6.16	17.4	1,223 1,147	64.7 53.4	83.5 73.4	10.1	
Shut........ Open........ Shut........ Open........	8.0 .6 8.2 .6	7.8 .5 7.9 .5	1,100 1,125+ 1,100 1,120	Open.. Shut..	1,122 1,101	.55 8.02	7.47	21	1.87	5.5	1,225 1,202	65.0 61.5	83.7 80.6	3.1	
Shut........ Open........ Shut........ Open........	5.5 .6 5.8 .6	5.3 .5 5.6 .5	1,110– 1,120– 1,110– 1,120–	Open.. Shut..	1,118 1,108	.55 5.55	5.00	10	.894	2.7	1,220 1,209	64.2 62.5	83.1 81.6	1.5	
Shut........ Open........ Shut........ Open........	4.2 .6 4.3 .6	4.0 .5 4.1 .5	1,115 1,125 1,115 1,120	Open.. Shut..	1,123 1,115	.55 4.15	3.60	8	.713	2.1	1,226 1,217	65.2 63.7	83.8 82.6	1.2	
Shut........ Open........ Shut........ Open........	2.1 .6 2.1 .6	2.0 .5 2.0 .5	1,115+ 1,120 1,115+ 1,120	Open.. Shut..	1,120 1,117	.55 2.05	1.50	8	.268	.6	1,223 1,219	64.7 64.1	83.5 82.9	.6	
Shut........ Open........ Shut........ Open........ Shut........ Open........	8.6 .5 9.2 .6 9.2 .6	8.3 .4 8.8 .5 8.7 .5	1,100 1,135 1,110 1,135 1,110– 1,135	Open.. Shut..	1,135 1,106	.50 8.80	8.3	29	2.55	7.4	1,239 1,207	67.3 62.2	85.6 81.2	4.4	
Shut........ Open........ Shut........ Open........	7.0 .5 7.3 .6	6.6 .5 7.0 .5	1,115 1,135 1,115 1,140	Open.. Shut..	1,138 1,115	.53 7.23	6.7	23	2.02	6.0	1,242 1,217	67.7 63.8	86.0 82.6	3.4	
Shut........ Open........ Shut........ Open........	6.2 .6 6.3 .6	5.9 .5 5.9 .5	1,125 1,140 1,125 1,140	Open.. Shut..	1,140 1,125	.55 6.08	5.53	15	1.32	3.9	1,245 1,228	68.3 65.5	86.4 84.1	2.3	
Shut........ Open........ Shut........ Open........	4.2 .6 4.2 .6	3.9 .5 4.0 .5	1,130– 1,140– 1,130 1,140	Open.. Shut..	1,139 1,129	.55 4.08	3.53	10	.88	2.6	1,243 1,233	67.9 66.3	86.1 84.7	1.4	

TABLE I.—*Runs at constant throttle, varying back pressures*—Continued.

Observations.				Results. (Based on average of observations.)											
Position of end outlet valve.	Back pressure (inches of mercury).		Tachometer reading (r.p.m.)	Position of end outlet valve.	Average tachometer (r.p.m.)	Average back pressure (in. mercury).	Applied back pressure (in. mercury).	Speed change, not corrected (r.p.m.)	Percent speed change (max. r.p.m. =100 per ct.)	Percent power loss.	True speed (r.p.m.)	Brake horsepower.	Brake M. E. P. (pounds per square inch).	Loss of brake M. E. P. due to applied back pressure.	
	South side.	North side.													
Shut	2.0	1.9	1,135												Aug. 29, 1916
Open	.6	.5	1,140	Open	1,141	0.55	1.40	6	0.53	1.6	1,246	68.4	86.5	.9	
Shut	2.0	1.9	1,135	Shut	1,135	1.95					1,239	67.3	85.6		
Open	.6	.5	1,140+												
Shut	3.2	3.0	1,135												
Open	.6	.5	1,140	Open	1,139	.55	2.37	4	.352	1.0	1,243	67.9	86.2	.6	
Shut	3.1	2.4	1,135	Shut	1,135	2.92					1,239	67.3	85.6		
Oepn	.6	.5	1,135+												
Shut	4.3	4.2	1,135												
Open	.6	.5	1,140												
Shut	4.3	4.2	1,125												
Open	.55	.5	1,135	Open	1,137½	.54	3.71	12½	1.10	3.2	1,241	67.6	85.9	1.8	
Shut	4.3	4.2	1,115	Shut	1,125	4.25					1,228	65.5	84.1		
Shut	3.7	3.6	1,115												
Open	.5	.4	1,125												
Shut	3.9	3.8	1,115	Open	1,127—	.49	3.36	12	1.06	3.2	1,230	65.9	84.4	1.8	
Open	.5	.5	1,130	Shut	1,115	3.85					1,217	63.7	82.6		
Shut	3.9	3.8	1,115												
Open	.55	.5	1,125												
Shut	6.9	6.7	1,110—												
Open	.6	.5	1,130+	Open	1,127	.56	6.29	20	1.77	5.2	1,230	65.9	84.4	3.0	
Shut	7.0	6.8	1,105+	Shut	1,107	6.85					1,208	62.4	81.4		
Open	.6	.55	1,125+												
Shut	9.8	9.6	1,080												
Open	.6	.5	1,100+												
Shut	10.3	10.2	1,080	Open	1,109	.57	10.23	26	2.34	6.8	1,210	62.7	81.7	3.8	
Open	.6	.5	1,110	Shut	1,083	10.80					1,182	58.4	77.9		
Shut	10.4	10.2	1,090												
Open	.65	.55	1,115												
Shut	12.1	11.9	1,070												
Open	.65	.55	1,125	Open	1,123	.6	11.42	53	4.72	12.9	1,226	65.2	83.8	7.7	
Shut	12.1	12.0	1,070	Shut	1,070	12.02					1,168	56.3	76.1		
Open	.65	.55	1,120												
Open	.65	.45	1,125												
Shut	12.1	11.8	1,085+	Open	1,135	.59	11.48	49	4.31	12.4	1,239	67.3	85.6	7.3	
Open	.65	.50	1,140	Shut	1,086	12.07					1,185	58.8	78.3		
Shut	12.3	12.1	1,085												
Open	.7	.6	1,140												

TABLE II.—*Runs at varying throttle, back pressure conditions fixed.*

Back pressure conditions fixed, in the sense that the obstructions to the escape of exhaust gas were unchanged through the series of runs However, the back pressure changed with engine power output due to change of amount of exhaust gas.

End outlet valve alternately open and closed, relieving or applying the back pressure.

Power output varied by changing the throttle position. These runs correspond to throttling an engine loaded by propeller with the exhaust escaping through a given manifold and muffler.

Data of Aug. 24, 1916, was obtained by forcing (by closing the end outlet valve) the exhaust to leak through the walls of a 4-foot length of 2½ inches flexible metallic exhaust hose and the joints of the connections.

Data of Aug. 29, 1916, was a similar set up with fewer leaks.

Observations.				Results. (Based on average of observations.)											
Position of end outlet valve.	Back pressure (inches of mercury).		Tachometer reading (R. P. M.).	Position of end outlet valve.	Average tachometer (R. P. M.).	Average back pressure (in. mercury).	Applied back pressure (in. mercury).	Speed change not corrected (R. P. M.).	Per cent speed change.	Per cent power loss.	True speed (R. P. M.).	Brake horsepower.	Brake M.E.P. (pounds per square inch).	Loss of brake M.E.P. (pounds per square inch).	
	South side.	North side.													
Shut	9.8	9.5	1,075												Aug. 24, 1916
Open	.55	.45	1,115	Open	1,115	0.53	9.30	43	3.81	11.0	1,217	63.8	82.5	6.2	
Shut	10.2	9.8	1,070	Shut	1,072	9.83					1,170	56.7	76.3		
Open	.6	.5	1,115												
Open	.35	.3	1,040												
Shut	7.00	6.6	1,000—	Open	1,042	.30	6.5	42	4.08	11.8	1,138	52.1	72.2	5.7	
Open	.30	.25	1,045	Shut	1,000	6.8					1,092	46.1	66.5		
Shut	6.90	6.7	1,000+												
Open	.15	.1	875												
Shut	3.6	3.6	850—	Open	875	.125	3.48	25	2.86	8.4	955	80.8	50.9	2.8	
Open	.15	.1	875	Shut	850	3.6					928	28.3	48.1		
Shut	3.6	3.6	850												
Open	.1	0	345	Open	345	.05	.175	0	0	0	377	1.9	7.9	0	
Shut	.25	.2	345	Shut	345	.225					377	1.9—	7.9		

TABLE II.—*Runs at varying throttles, back pressure conditions fixed*—Continued.

Observations.				Results. (Based on observations.)											
Position of end outlet valve.	Back pressure (inches of mercury). South side.	Back pressure (inches of mercury). North side.	Tachometer reading (R. P. M.).	Position of end outlet valve.	Average tachometer R. P. M.).	Average back pressure (in. mercury).	Applied back pressure (in. mercury).	Speed change not corrected (R. P. M.).	Per cent speed change.	Per cent power loss.	True speed (R. P. M.).	Brake horsepower.	Brake M. E. P. (pounds per square inch).	Loss of brake M. E. P. (pounds per square inch).	
Open	.05	0	525	Open	530	.025	0.51	0	0	0	578	6.9	18.7	0.1	
Shut	.6	.5	530	Shut	530	0.54					578	6.8	18.6		
Open	.05	0	530												
Shut	.55	.5	530+												
Open	.05	0	610−	Open	610	.025	.71	± 3	0.5	1.5	665	10.4	24.7	0.3	Aug. 24, 1916
Shut	.8	.7	605+	Shut	605+	.74					660	10.2	24.4		
Open	.05	0	610−												
Shut	.75	.7	605+												
Open	.4	.3	1,075												
Shut	9.2	8.8	1,050												
Open	.5	.4	1,110−												
Shut	9.6	9.4	1,060	Open	1,112	0.50	9.34	44	3.96	11.4	1,214	63.4	82.2	6.4	
Open	.7	.6	1,130	Shut	1,068	9.84					1,166	56.1	75.8		
Shut	10.6	10.4	1,080												
Open	.7	.6	1,135												
Shut	10.5	10.2	1,080+												
Shut	12.1	11.9	1,070												Aug. 29, 1916
Open	.65	.55	1,125	Open	1,122+	.6	11.4	+ 52	4.67	13.4	1,126	65.2	83.8	7.6	
Shut	12.1	12.0	1,070	Shut	1,070	12.0					1,169	56.5	76.2		
Open	.65	.55	1,120												
Shut	8.7	8.7	995												
Open	.3	.35	1,080												
Shut	8.7	8.6	995	Open	1,073	.33	8.34	78	7.27	20.3	1,171	56.8	76.5	10.7	
Open	.3	.35	1,060	Shut	995	8.67					1,086	45.3	65.8		
Shut	8.7	8.6	995												
Open	.35	.35	1,080												
Open	.15	.25	1,000	Open	998	.20	6.68	60	6.01	17.0	1,089	45.7	66.2	7.7	
Shut	6.8	6.8	940	Shut	938	6.88					1,024	38.0	58.5		
Open	.15	.25	1,000−												
Shut	6.8	6.7	940−												
Shut	4.3	4.3	830	Open	860	.10	4.25	30	3.49	10.1	939	29.3	49.2	3.4	
Open	.10	.1	860	Shut	830	4.35					906	26.3	45.8		
Shut	4.4	4.4	830												
Open	.1	.1	860												
Open	.10	.05	695	Open	694	.08	1.72	14	1.98	5.8	758	15.4	32.1	1.4	
Shut	1.80	1.80	680	Shut	680	1.78					742	14.5	30.7		
Open	.05	.05	690+												
Shut	1.80	1.70	680												
Shut	12.1	11.8	1,085+	Open	1,140	.60	11.47	54	4.74	13.5	1,244	68.1	86.4	8.0	
Open	.65	.50	1,140	Shut	1,086	12.07					1,186	59.0	78.4		
Shut	12.3	12.1	1,085												
Open	.7	.6	1,140												

TABLE III.—*Run with and without wire mesh muffler, varying throttle position, other conditions unchanged.*

[Runs of Aug. 29, 1916.]

Outlet conditions.	Back pressure (inches of mercury). South side (farthest from outlet).	Back pressure (inches of mercury). North side (nearest outlet).	Tachometer reading (R.P.M.).	Average back pressure (in. mercury).	True speed (R.P.M.).	Brake horsepower.
Muffler in position beyond quick-opening valve	0.02	0.02	730	0.02	797	17.9
	.03	.03	830+	.03	908	26.5
	.10	.10	930	.10	1,014	36.9
	.10	.20	1,020	.15	1,113	48.8
	.30	.35	1,100−	+.32	1,198	60.8
	.45	.45	1,140	.45	1,244	68.1
	.75	.65	1,170	.70	1,276	73.4
Without muffler	0	0	610+	0	663	10.6
	.02	.02	740	.02	808	18.7
	.07	.07	890	.07	971	32.4
	.10	.20	980	.15	1,069	43.2
	.25	.30	1,090	.275	1,190	59.6
	.40	.45	1,145	.425	1,250	69.1

TABLE IV.—*G. P. F. Mufflers; Throttle Varied.*

Mufflers placed one on the end of each straight exhaust manifold for each block of four cylinders of the Curtiss engine. One pair of the 28-inch long mufflers tested first, then the pair of 12-inch long mufflers were used.

[Sept. 8, 1916.]

TWO 28-INCH G. P. F. MUFFLERS.

Observations.					Results.										
Throttle position (hole No.).	Muffler on or off.	Back pressure (inches of mercury). South side.	Back pressure (inches of mercury). North side	Tachometer reading (R.P.M.).	Muffler on or off.	Average tachometer (R. P. M.).	Average back pressure (in. mercury).	Back pressure due to muffler (in. mercury).	Speed change due to muffler (R. P. M. not corrected).	Per cent speed change due to muffler.	Per cent power loss due to muffler.	True speed (R. P. M.).	Brake horsepower	Brake M. E. P. (pounds per square inch).	Loss of brake M. E. P. due to muffler.
4	Off......	0.05	0	840	Off......	850	0.01	0.63	0	0	0	928	28.3	48.1	0
	On......	.6	0.06	850	On......	850	.64					928	28.3	48.1	
	Off......	0	0	860											
	On......	.7	.65	850											
5	Off......	0	0	970	Off......	977½	.025	1.475	5	0.512	1.5	1,066	42.8	63.4	0.7
	On......	1.5	1.50	980	On......	972½	1.5					1,060	42.1	62.7	
	Off......	.05	.05	975											
	On......	1.5	1.5	975											
8	Off......	0	0	1,160	Off......	1,162½	0	4.0	26	2.24	6.6	1,268	72.1	89.7	3.9
	On......	4.0	4.0	1,140	On......	1,136½	4.0					1,240	67.4	85.8	
	Off......	0	0	1,165											
	On......	4.0	4.0	1,130+											
TWO 12-INCH G. P. F. MUFFLERS.															
4	Off......	0	0	850	Off......	855	0	0.25	0	0	0	933	28.8	48.6	0
	On......	0.25	0.25	855	On......	855	0.25					933	28.8	48.6	
	Off......	0	0	855											
	On......	.25	.25	855											
5	Off......	.05	0	980+	Off......	982	0	.55	0	0	0	1,071	43.4	64.0	0
	On......	.55	.55	980+	On......	982	0.55					1,071	43.4	64.0	
	Off......	0	0	980+											
	On......	.55	.55	980+											
[1]8	Off......	0	0	1,150+	Off......	1,149	0	1.9	6	0.522	1.5	1,254	69.8	87.7	0.9
	On......	1.6	2.1	1,150−	On......	1,143	1.9					1,247	68.6	86.8	
	Off......	0	0	1,145											
	On......	1.7	2.1	1,135+											

[1] Position for 75 H. P. as used in Table I.

TABLE V.—*Muffler tests on marine type, 4-cylinder two-cycle engine.*

B. H. P. of engine on these tests about 25. Exhaust noise about that of a 75 H. P. 8-cylinder 4-cycle airplane engine when exhaust was open.

Muffler.	Back pressure (inches mercury). With.	Back pressure (inches mercury). Without.	Back pressure (inches mercury). Due to muffler.	Speed (R. P. M.).	Remarks.
Variable pitch spiral	1.0±	0.75±	0.25	1,050-1,200	Noise over one-half stopped; exhaust still cracks a little. Back pressure very small; weight of muffler large.
One unit, "manifold muffler,"[1] end blocked; gas goes around end of inside cylinder.	1.30	.80	.50	1,200	Deadens sound more than spiral did.
One unit, "manifold muffler,"[1] end blocked; gas goes around end of inside cylinder.	....	....	.30	1,230	Muffler at end of long pipe through window to outdoors. Longer manifold before muffler may lower back pressure of muffler, but adds back pressure of its own that may be worse.
Two units of manifold muffler connected as designed.	....	....	.50	1,225	Back pressure same from either part of double set-up, showing that it makes little difference whether gas goes around end of cylinder or not. Observed noise from distance of one-fourth mile, in direction to which open exhaust pointed. Noise strong when muffler was off, very like an airplane passing close overhead. Noise practically, but not quite, inaudible with muffler on. One could notice there was an engine somewhere, but the noise would not locate it.

[1] Weight of one unit of "manifold muffler" about 4 pounds.

TABLE VI.

Muffler tests on Curtiss OX 8-cylinder, 4 by 5 inch motor, using whirl chamber mufflers. A check run on manifold muffler.
Notes on ability of mufflers to silence the crack of the exhaust taken on Ingeco single 5.5 by 10 inch cylinder farm engine, loaded and with retarded spark.

[Apr. 10, 1917.]

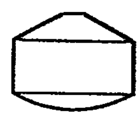
Muffler set-up, for all 8 cylinders.				Muffler off or on.	Back pressure (inches mercury).		Tachometer reading (r.p.m.)	True speed (r.p.m.)	Back pressure due to muffler.	Per cent power loss, from B. P. power loss curves.	Crack or bark suppression on Ingeco; general silencing ability; and remarks. (Fractions are reduction of noise, averages of estimates by 2 or more observers. Sketches show cross section of whirl chamber arrangements.)
Diameter of whirl chamber ring (inches).	Position of cone.	Position of cover plate.	Width of nozzle (inches).		South side.	North side.					
12	In	In	1	Off On On	0.5 1.6 1.6	0.10 1.5 1.4	1,140 1,140 1,140	1,245	1.2	1.0	½+off crack.
7	In	In	1	Off On On	.5 1.8 1.9	.1 1.7 1.8	1,130 1,130 1,130	1,234	1.5	1.2	½±.
7	In	In	.5	Off On	.3 1.9	0 1.65	1,110 1,110	1,212	1.6	1.3	¾. 
4	In	In	1	Off Off Off Off On On	.20 .25 .30 .40 1.8 1.9	.05 .05 .05 .05 1.6 1.6	1,100 1,100 1,100 1,100 1,100 1,100	1,201	1.55	1.25	½±.
12	In	Out	1	Off On	.40 1.40	.10 1.20	1,140 1,140	1,245	1.05	.85	½+. ¼+.
7	In	Out	1	Off On	.40 1.45	.05 1.2	1,130 1,130	1,235	1.10	.90	½. 
7	In	Out	.5	Off On	.30 1.7	0 1.5	1,140 1,140	1,245	1.45	1.15	½+.
4	In	Out	1	Off On On	.30 1.2 1.3	.05 1.0 1.15	1,120 1,120 1,120	1,223	1.0	.83	¼—, or less.
12	Out	Out	1	Off On	.3 1.35	.2 1.10	1,130 1,130	1,235	—1.0	.8	⅔.
7	Out	Out	1	Off On	.4 1.3	.05 1.1	1,110 1,110	1,212	—1.0	.8	½. 
7	Out	Out	.5	Off On	.3 1.7	0 1.6	1,190 1,190	1,300	1.45	1.15	¾; good muffling.
4	Out	Out	1	Off On	.3 1.05	.05 .80	1,100 1,100	1,201	.75	.63	Barks and whistles; bad.
12	Removed.	Removed.	1	Off On	.5 .6	.05 .25	1,130 1,130	1,234	.15	.12	
7	Removed.	Removed.	1	Off On	.5 .6	.05 .25	1,100 1,100	1,201	.15	.12	
7	Removed.	Removed.	.5	Off On	.5 .75	.05 .35	1,090 1,090	1,190	.27	.23	
4	Removed.	Removed.	1	Off On	.45 .50	.05 .30	1,050 1,050	1,147	.15	.12	
Manifold muffler, one unit, as designed				Off On On Off	.5 1.2 1.35 .5	.1 1.0 1.10 0	1,140+ 1,140+ 1,110 1,110	1,247 1,212	.8 1.0	.65 .83	½. Good muffling.

Table VII.

Tests of mufflers, continuing the investigations of capacity. First half of tests, mufflers taking exhaust from all 8 cylinders of Curtiss OX engine Second half of tests, mufflers taking exhaust from only 1 clock, 4 cylinders, north side of engine. Throttle position No. 8.

[Apr. 21, 1917.]

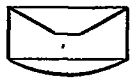
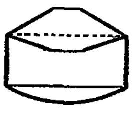
Muffler and set-up, specifications of whirl-chamber mufflers.				Back pressure (inches of mercury).				Tachometer (multiply by 1.092 for true R. P. M.).		Back pressure due to muffler.	Remarks; sketches of whirl. chamber sections.
				Muffler off.		Muffler on.					
Ring diameter.	Cone.	Cover.	Nozzle width.	South.	North.	South.	North.	Off.	On.		
											ONE MUFFLER ON 8 CYLINDERS.
12	In	Out	1	0.10?	0.05	1.60	1.4	1,150	1,160	1.2	
12	In	Out	1	.4	.10	1.55	1.4	1,160	1,155	1.2	
7	In	Out	1	.40	.05	1.5	1.3	1,160	1,160	.9	
4	In	Out	1	.45	.10	1.3	1.1	1,160	1,155	.65	
Manifold muffler				.4	.1	1.0	.8	1,160	1,150		
											ONE MUFFLER ON 4 CYLINDERS.
12	In	Out	1		0.05		0.6	1,140	1,140	0.55	
7	Shallow cone in.	Out	1		.05		.9	1,140	1,140	.85	Shallow cone fails to muffle Ingeco=¼ off bark. 
4	In	Out	1		.05		.7	1,140	1,140	.65	
Manifold muffler					.05		.7	1,150	1,150	.65	
7	Shallow = In. Regular = Out Diffuser plate between.	Out	1		.05		1.0	1,150	1,150	.95	Ingeco muffling fair=½+; diffuser plate=sheet metal perforated with about 100 ⅛-inch holes. 

Table VIII.—*Tests of muffling qualities.*

Exhaust carried outdoors by connecting to the end of cross manifold two 6-foot lengths of 2-inch standard wrought-iron pipe, joined by a 45-inch cast-iron ell. This pipe caused 1 inch of mercury back pressure, additional, on the engine, but did not silence the exhaust noises. Curtiss engine. Throttle position No. 8.

The values of noise suppression were obtained from the collaborated estimates of a number of observers. One stationed in a direct line with the end of the exhaust pipe. Another at right angles to the line of the pipe, in the plane of the muffler. Another at the end of the pipe changing mufflers, and perhaps others scattered around. The first two were generally about 50 feet away from muffler, but other distances were also used when muffling qualities of two devices were nearly the same.

APR. 21, 1917.

Muffler and description.	Back pressure (inches mercury).				Tachometer (multiply by 1.092 for true R. P. M.).		Back pressure due to muffler.	Remarks, estimated bark suppression, in fraction or per cent.
	Muffler off.		Muffler on.					
	South.	North.	South.	North.	Off.	On.		
Manifold muffler, inside cylinder not touching end plate.	0.8	0.7	1.5	1.4		1,150	0.7	½; steady swish, no crack.
12-inch whirl, cone in, cover out	1.0	.8	1.6	1.3		1,150	.55	¾; steady swish, less hum than manifold.
7 by 1 inch whirl, shallow cone in, deep cone out, no diffuser.	1.2	.8	1.8	1.7		1,150	.75	⅝; sharper crack.
4-inch whirl, cone in, cover out	1.2	1.0	1.5	1.5		1,160	.4	¼; very distinct crack.
Manifold muffler, end of inner tube closed,	1.2	1.0	1.9	1.7		1,160	.7	⅝; can distinguish separate exhausts.
exhuast escapes only through holes in the tube (as designed).	.8	.6	1.4	1.5		1,160	.75	⅞; swish of explosions, hums steadily.
12-inch whirl, cone out, cover out	1.1	.9	1.5	1.4		1,165	.45	¾; more exhaust noise.
7 by 1 inch whirl, deep cone out, cover out	1.2	.9	1.8	1.6		1,165	.65	⅝; still more noise.
Manifold muffler, same as last	1.2	1.0	1.9	1.7		1,170	.7	⅞+, the loud cylinders scarcely audible.
MAY 12, 1917.								
Manifold muffler, tube end closed	0.9	0.8	1.6	1.6	1,170	1,170	0.75	80 per cent, light swish and hum.
G. P. F. 12-inch	1.1	1.0	3.2	3.0	1,170	1,165	2.05	80 per cent+; sharp swish.
7 by 1 inch whirl, venturi substituted for cone, cover out.	1.2	1.0	2.0	1.8	1,170	1,170	.8	50 per cent; metallic bark.
Manifold muffler, tube end closed	1.1	.8	1.85	1.7	1,140	1,140	.8	The different character of sound from G. P. F. and M. M. is hard to compare. Both are successful silencers.
G. P. F. 12-inch	1.25	1.0	2.9	2.7	1,135	1,130	1.7	
Manifold muffler, as last	1.05	.85	1.7	1.45	1,120	1,130	.6	
G. P. F. 12-inch	1.25	1.0	2.95	2.6	1,130+	1,130−	1.8	
Venturi (only)	1.25	1.0	1.6	1.55	1,130	1,130	.5 +	Unearthly howl; loud shriek.
Manifold muffler, as last	1.2	1.0	1.8	1.6	1,145	1,145	.6	
Shell of manifold muffler, inner tube removed.	1.2	1.0	1.9	1.6	1,145	1,145	.65	Roars and whistles.

TABLE VIII.—*Tests of muffling qualities*—Continued.

MAY 14, 1917.

Muffler and description.	Back pressure (inches mercury).				Tachometer (multiply by 1.092 for true R. P. M.).		Back pressure due to muffler.	Remarks.
	Muffler off.		Muffler on.					
	South.	North.	South.	North.	Off.	On.		
Duplex whirl, 1-inch entering nozzle	0.9	0.7	1.4	1.8	1,110	1,110		General conclusions from tests of May 14, after repeated trials, were that the Duplex is slightly the better silencer than the manifold; both very good.
			1.3	1.1	......	1,110	0.5	
Manifold muffler, regular	1.0	.8	1.6	1.3	......	1,130	.6 —	
Duplex whirl, 1-inch entering nozzle	......	......	1.5	1.4	......	1,110	.6	
Do	1.1	.9	1.7	1.6	1,100	1,100	.65	
Manifold muffler, regular	1.3	1.0	1.8	1.7	1,100	1,100	.60	
Duplex whirl, ½-inch entering nozzle	1.3	1.0	1.8	1.7	1,110	1,110	.60	
Manifold muffler, regular	1.3	1.0	1.9	1.7	1,105	1,100	.65	
Manifold muffler, regular	1.2	......	1.8	......	[1]1,100	[1]1,100	.6	
Duplex, whirl 1-inch entering nozzle	1.2	......	1.8	......	[1]1,100	[1]1,100	.6	
Duplex whirl, ½-inch entering nozzle	1.2	......	1.8	......	[1]1,100	[1]1,100	.6	
Manifold muffler, regular	1.2	......	1.8	......	[1]1,100	[1]1,100	.6	

MAY 16, 1917.

Muffler and description.	Muffler off, South.	Muffler off, North.	Muffler on, South.	Muffler on, North.	Tachometer, Off.	Tachometer, On.	Back pressure due to muffler.	Remarks.
Duplex whirl	1.0	0.75	1.6	1.4	1,125	1,125	0.65	Duplex whirl and long manifold muffler about equal in silencing; both good.
Long manifold muffler	1.1	.95	1.5	1.3	1,130	1,125	0.4	
Duplex whirl	1.2	1.0	1.7	1.5	1,125	1,125	.5	
Long manifold muffler	1.2	1.0	1.6	1.6	1,125	1,120	.5	

[1] About.

TABLE IX.—*Tests of muffling qualities.*

Various mufflers used. Otherwise apparatus same as that described in Table VIII. Tests made May 19 and 25, 1917, on Curtiss engine Tests on Ingeco engine for bark suppression made on May 25, are shown in "Remarks."

Adam C. Davis, jr., C. A. Pierce, G. M. Rogers, H. Diederichs, V. R. Gage, G. B. Upton, and Birton N. Wilson individually made notes, and afterwards collectively made the final decision on the silencing qualities of the various devices.

MAY 19, 1917.

Muffler and description.	Back pressure (inches mercury).				Tachometer (multiply by 1.092 for true R. P. M.).		Back pressure due to muffler.	Remarks.	Weight (pounds).
	Muffler off.		Muffler on.						
	South.	North.	South.	North.	Off.	On.			
Duplex whirl, uniformly perforated	1.0	0.8	1.4	1.2	1,030	1,050+	......	These 3 mufflers all good silencers, G. P. F. judged best in respect to noise suppression.	
diffuser, as before	1.2	1.0	1.5	1.3	1,060	1,060	0.35		3.0
Long manifold muffler	1.3	1.0	1.5	1.3	1,060	1,060	.25		4.3—
G. P. F. 12-inch	1.2	1.0	2.8	2.8	1,060	1,060	.70		4.3
G. P. F. 12-inch	......	......	2.9	2.8	1,065	......	1.75		
12-inch whirl	1.3	1.0	1.5	1.4	1,060	1,060	.3	Slight roar, perhaps due to the light material used in construction.	3.4
Manifold muffler, regular	.9	.7	1.5	1.2	1,050	1,060	.5	Second best on swish; third on hum	4.6
Duplex whirl, graduated diffuser	1.1	.8	1.4	1.2	1,060	1,055	.35	First best on swish; second on hum	3.0
G. P. F. 12-inch	1.1	.8	2.2	2.0	1,045	1,040	1.2	Third best on swish; first on hum	4.3
Max m	1.1	.8	2.9	2.8	1,110	1,110	1.9	Of these 3, Maxim judged best silencer, manifold muffler next, and G. P. F. last; all very good.	12.0
G. P. F. 12-inch	1.1	.8	2.3	2.0	1,125	1,110	1.2		4.3
Manifold muffler, regular	1.1	.8	1.5	1.3	1,120	1,120	.45		4.6

MAY 25, 1917.

Muffler and description.	Muffler off, South.	Muffler off, North.	Muffler on, South.	Muffler on, North.	Tachometer, Off.	Tachometer, On.	Back pressure due to muffler.	Remarks.	Weight (pounds).
New manifold, double inner tube	0.8	0.5	1.4	......	1,140	1,140	0.4	New manifold muffler is much better silencer than G. P. F.	7.2
G. P. F. 12-inch	1.0	......	2.0	......	......	1,130	1.0		4.3
	1.0								
New manifold, single inner tube	1.0	......	1.35	......	1,145	1,145	.35	New manifold barks more without second inner tube. Now equals G. P. F. as a silencer. A short tail pipe on new manifold muffler does not change back pressure or silencing.	5.9
G. P. F. 12-inch	.9	......	1.8	......	1,140	1,140	.9		4.3

On Ingeco engine the new manifold muffler with two perforated inner tubes is easily better than G. P. F. 12-inch or the duplex whirl chamber mufflers.

TABLE X.—*Showing how the back pressure due to any one muffler apparently decreased as time passed. This effect is due to increasing leakage in the exhaust piping system.*

G=G. P. F. 12" Muffler.
M="Manifold Muffler" (regular).
L="Long Manifold."
D="Duplex Whirl Chamber."
12="Whirl Chamber," 12".
7="Whirl Chamber," 7" x 1".

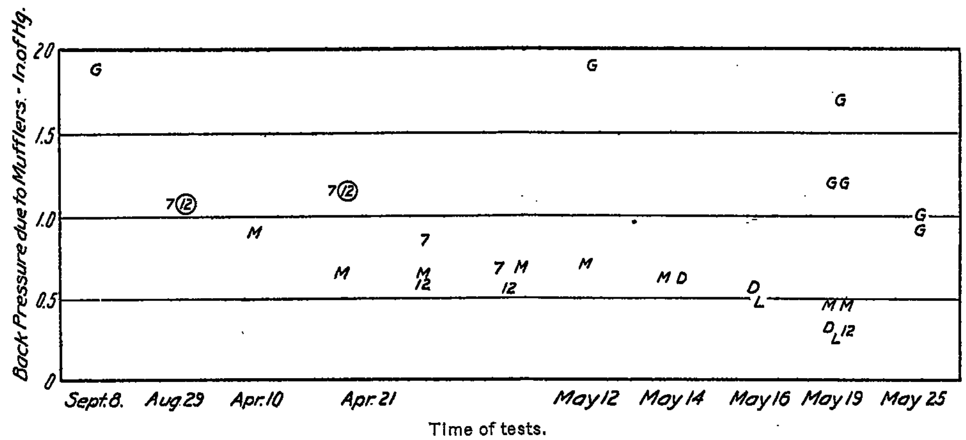


TABLE XI.—*Summary of results.*

Muffler.	Weight (pounds).	Curtiss, 8-cylinder, 70-horsepower. Back pressure (inches of mercury).	Curtiss, 8-cylinder, 70-horsepower. Power loss (per cent).	Noise suppression estimated in per cent).	Silencing qualities.
G. P. F., 12-inch	4.3	2.0	1.6	75	Very good.
G. P. F., 28-inch	....	4.0	3.2	40–70	Good minus.
Maxim (Ford)	12.0	3.6	2.9	75–80	Very good plus.
Manifold type	4.6	.8	.7	75	Very good.
Long manifold	....	.7	.6	....	Verygood.
New manifold:					
2 inner tubes	7.2	.8	.7	75–80	Very good plus.
1 inner tube	5.9	.8	.7	75	Very good.
Whirl chamber type:					
4-inch	1.2	.7	.6	30–40	Fair minus.
7 by ½ inch	1.8	1.0	.8	50–60	Fair plus.
7 by 1 inch	1.7	.8	.7	50	Fair plus.
12-inch	3.4	.8–	.6–	50–70	Fair plus.
Duplex	3.0	.8	.7	....	Good plus.
Wire mesh	15.0	.3(?)	.2(?)	50	
Flexible metallic exhaust hose, 2 inches diameter	[1] 1	....	....	....	
Curtiss stock manifold; 2 required	[2] 5.5	....	....	....	

[1] Per foot of length.
[2] Each.

APPENDIX A.

FAN DYNAMOMETER DESIGN AND CALIBRATION.

Because of its flexibility and inherent regulation, due to torque varying as square of speed, and similarity to a propeller, the dynamometer chosen was the fan type. The general scheme of design is shown in figure 23. The resistance plates *PP* are of Tobin bronze plate $\frac{3}{16}$ inch thick. The length, a, of each plate parallel to the shaft is 14 inches. The radial width, b, of each plate is 10 inches. The plates are fastened to the steel arm, A, which revolves edgewise, by two angle irons riveted on the back of each plate, and bolted to the arm A. A series of evenly spaced holes in each end of arm A makes it possible to clamp the plates *PP* at any desired distances from the center, making the outside diameters of the plates, D_0, adjustable.

The fan shaft is mounted in ball bearings, which are supported by a framework built up of steel angle shapes. This frame is extended, as may be seen in the photograph, figure 1 of the main body of this report, to carry a rectangular box safety housing of wire mesh screen around the fan.

The design is copied in detail, both as to fan and framework, from a fan dynamometer at the A. C. A. testing laboratory in New York City, which in turn was built after the "Franklin" dynamometer. In each case the fan dynamometer of this design was tested out by driving

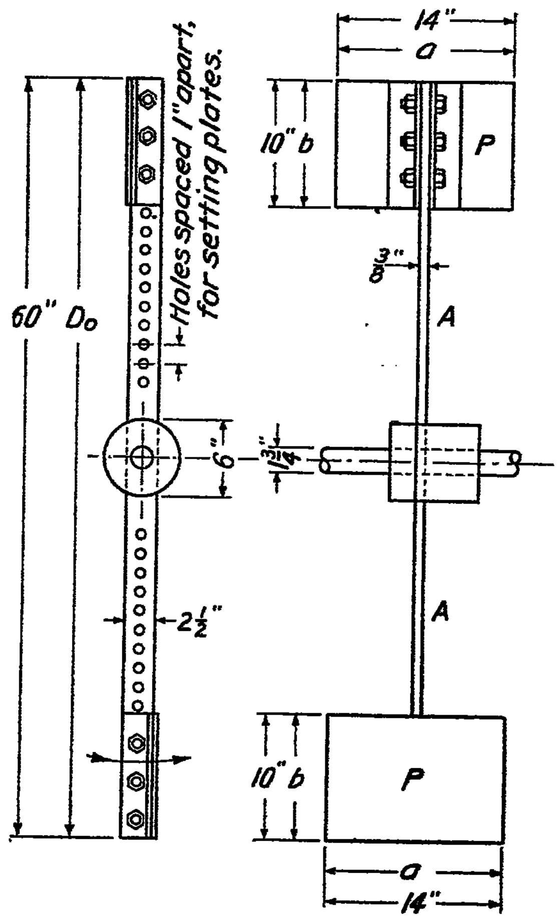


Fig. 23

from an engine or motor mounted on a cradle, and the calibration curves so obtained were available to us and used for our machine, as developed in the following analysis:

To check the accuracy of these calibrations we used the general formula for fan dynamometers, worked out by the White & Poppe Automobile Co., in England, and published in Automobile Engineer, August, 1910. The formula is quoted in the book on Dynamometers, by F. J. Jervis-Smith, on page 117. It is

$$HP = \frac{a^2 R^3 N^3}{4.01 \times 10^{15}}$$

in which a is the side dimension of a square plate, replacing the a and b of figure 23; $R = \frac{D_0}{2}$ of figure 23; and N = r. p. m. Dimensions were in centimeters in getting the constant 4.01×10^{15} above. We may generalize the formula into

$$HP = \frac{abD_0{}^3 N^3}{8 \times 4.01 \times 10^{15}}$$

for dimensions in centimeters, or

$$HP = \frac{2.54^5 abD_0{}^3 N^3}{8 \times 4.01 \times 10^{15}}$$

for dimensions in inches. This reduces to

$$HP = 3.43 \times 10^{-15} \times abD_0^3 \times N^3.$$

The formula is stated by White & Poppe to be valid only if there is no interference with in or out flow of air around the fan. If there is such interference, and less air is handled, the power is decreased below that given by the formula.

Taking the formula as

$$HP = K \times 10^{-15} \times abD_0^3 N^3,$$

we checked over the available calibrations of our dynamometer design by finding the values of K for different settings of D_0, and trying to account for variations by the expected interferences with air flow.

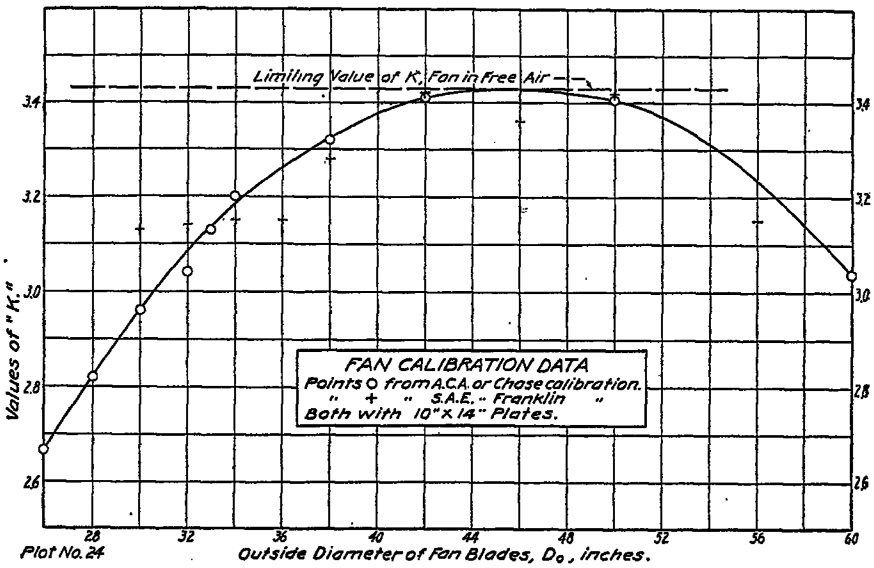
$$K = \frac{HP \times 10^{15}}{abD_0^3 N^3}$$

TABLE XII.—*S. A. E. or Franklin calibrations, 10 by 14 inch plates.*

Setting D_0	N	HP	K	Mean K
56	600	17.0	3.20	3.15
56	800	39.0	3.10	
50	800	30.7	3.43	3.42
50	1,000	60.0	3.42	
46	800	23.5	3.37	3.36
46	1,000	45.7	3.35	
42	1,000	35.5	3.42	3.42
42	1,200	61.5	3.42	
38	1,000	25.0	3.26	3.28
38	1,400	69.5	3.29	
36	1,000	20.7	3.17	3.15
36	1,400	56.2	3.14	
34	1,200	30.0	3.15	3.15
34	1,600	70.7	3.14	
32	1,200	25.0	3.15	3.14
32	1,600	59.0—	3.12	
30	1,200	20.0+	3.08	3.13
30	1,800	70.5	3.19	

TABLE XIII.—*A. C. A. calibration (Chase), 10 by 14 inch plates.*

Setting D_0	N	HP	K	Mean K
60	600	19.7	3.02	3.04
60	400	6.0	3.10	
50	700	20.4	3.40	3.41
50	500	7.5	3.43	
42	800	17.8	3.35	3.41
42	600	7.9	3.53	
38	900	18.1	3.23	3.32
38	600	5.8	3.49	
34	1,100	22.9	3.13	3.20
34	700	6.3	3.34	
33	1,100	20.8	3.11	3.13
33	800	8.2	3.18	
32	1,200	24.1	3.04	3.04
32	800	7.1	3.03	
30	1,300	24.5	2.95	2.96
30	900	8.2	2.98	
28	1,400	24.2	2.87	2.82
28	1,000	8.4	2.73	
26	1,600	27.1	2.69	2.67
26	1,100	8.6	2.63	



Plot No. 24

The relation of the K values to D_0, and of the two calibrations to each other, are shown in plot 24. The A. C. A. or Chase calibration seems probably the better, being notably more consistent internally. This may, however, be due to smoothing out of data by Chase, by cross-fairing methods. Low values of K for small values of D_0 are due to interference with intake air by the shielding framework around the fan; small values of K at large values of D_0 are due to interference of the floor and ends of framework with discharge air.

With given plate size and setting of D_0, we fix values of a, b, and D_0 of the formula $HP = K \times 10^{-15} a b D_0^3 N^3$, and K is also fixed by the setting, so that the formula becomes for any one setting $HP = \text{constant} \times N^3$. Picking values of K from plot 24 for the 10 by 14 inch plates, assuming the curve sketched in along the Chase points as correct, we have:

TABLE XIV.

D_0	K	Constant $=(K \times 10^{-15} \times 10 \times 14 \times D_0^3)$.
26	2.67	6.56×10^{-9}
28	2.82	8.66
30	2.96	11.2
32	3.08	14.1
34	3.19	17.6
36	3.26	21.3
38	3.32	25.5
40	3.38	30.2
42	3.41	35.4
44	3.42	40.8
46	3.42	46.6
48	3.42	52.9
50	3.41	59.6
52	3.37	66.2
54	3.31	72.8
56	3.23	79.3
58	3.15	85.9
60	3.04	91.8×10^{-9}

The "constant" gives the H.P. at 1,000 r.p.m. for each setting. The setting used throughout the muffler tests on the Curtiss engine was with $D_0 = 42$ inches, constant $= 35.4$ H.P. at 1,000 r.p.m.

APPENDIX B.

TESTS OF AUTOMOBILE ENGINE MUFFLERS AT UNIVERSITY OF MICHIGAN.

[Reported in Horseless Age, May, 1915.]

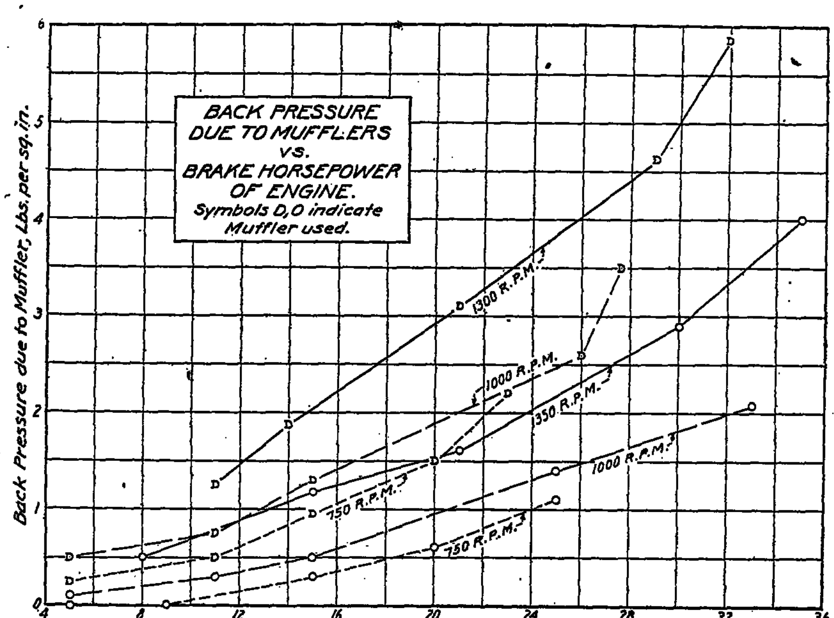
These tests were made on a stock Hudson 6-cylinder motor of 4⅛-inch bore by 5¼-inch stroke. For the purposes for which we would like to use the data the method of test was unfortunate. Runs were made through the range of throttle positions and speeds without a muffler, followed by similar sets of runs with the muffler tested. Back pressures were read with the mufflers. Power losses were inferred by comparison of the runs with and without the mufflers. As these runs were somewhat separated in time, power changes due to changes in carburetion, lubrication, ignition, etc., can not be sorted out from power changes due to the mufflers alone. The mufflers tested were commercial designs, five in number.

From the data we have sorted out parts from results on three of the mufflers (those found best as silencers), tabulating and plotting their results for our own information. The data are given in Table XV herewith. Inspection of the table, which may be taken to represent the state of the art of muffling at the date of 1915, shows how well grounded was the fear that the muffling of airplane engines would be accompanied with prohibitive loss of power. One commercial muffler lost about 18 per cent and another 14 per cent of the maximum engine power. At the same time, however, there was hope offered, in that the smallest and lightest muffler tested gave the best silencing and also the least power loss—only 3.6 per cent of the maximum engine power. This muffler weighed 14 pounds for a 40-horsepower motor, or 0.35 pounds per horsepower. We have now obtained, by contrast, excellent muffling on a 70-horsepower motor at a power loss less than 1 per cent and with a weight of 5 or 6 pounds (less than 0.09 pounds per horsepower).

TABLE XV.—*University of Michigan tests—Hudson 6, 4⅛ by 5¼ inches.*

Muffler.	750 r. p. m. B.H.P. with muffler.	750 r. p. m. Back pressure with muffler (pounds per square inch).	750 r. p. m. H. P. loss.	750 r. p. m. Per cent power loss.	750 r. p. m. Loss of brake M. E. P. (pounds per square inch).	1,000 r. p. m. B. H. P. with muffler.	1,000 r. p. m. Back pressure with muffler (pounds per square inch).	1,000 r. p. m. H. P. loss.	1,000 r. p. m. Per cent power loss.	1,000 r. p. m. Loss of brake M. E. P. (pounds per square inch).	1,300 r. p. m. B. P. H. with muffler.	1,300 r. p. m. Back pressure with muffler (pounds per square inch).	1,300 r. p. m. H. P. loss.	1,300 r. p. m. Per cent power loss.	1,300 r. p. m. Loss of brake M. E. P. (pounds per square inch).
D......	5	0.25	0.2	4.0	0.5	5	0.5	0.4	7.4	0.7	11	1.3	1.3	10.6	1.8
	11	0.5	0.2	1.8	0.5	11	0.75	0.5	4.3	0.9	14	1.9	1.8	11.4	2.5
	15	0.95	0.35	2.3	0.9	15	1.3	1.3	8.0	2.4	21	3.1	2.8	11.8	3.9
	20	1.5	1.0	4.8	2.5	26	2.6	1.6	5.8	3.0	29	4.6	4.6	13.7	6.5
	23	2.2	1.6	6.5	4.0	27.5	3.5	1.9	6.5	3.5	32	5.9	7.0	17.9	9.8
Y......	5.5	0	0.1	1.8	0.25	5	0	0.05	1.0	0.1	10	0.2	0.05	0.5	0.07
	10.5	0	0.1	0.9	0.25	10	0.1	0.3	2.9	0.55	16	0.45	0.05	0.3	0.07
	16	0.2	0.3	1.8	0.75	15	0.2	0.3	2.0	0.55	19	0.6	0.5	2.6	0.7
	20	0.25	0.3	1.5	0.75	24	0.5	0.5	2.0	0.9	29	1.0	1.0	3.4	1.4
	24	0.5	0.9	3.6	2.3	31	0.9	0.6	1.9	1.1	38	1.1	1.4	3.6	2.0
O......	5	0	0.1	2.0	0.25	5	0.1	0	2.0	0	[1] 8	0.5	0	0	0
	9	0	0.2	2.2	0.5	11	0.3	0.5	2.6	0.9	[1] 15	1.2	0.8	5.1	1.1
	15	0.3	0.1	0.7	0.25	15	0.5	0.8	5.1	1.5	[1] 21	1.6	2.5	10.6	3.4
	20	0.6	0.1	0.5	0.25	25	1.4	1.8	6.7	3.3	[1] 30	2.9	3.6	10.7	4.9
	25	1.1	0.6	2.3	1.5	33	2.1	0.9	2.7	1.7	[1] 35	4.0	5.6	13.8	7.6

[1] This group at 1,350 r. p. m.



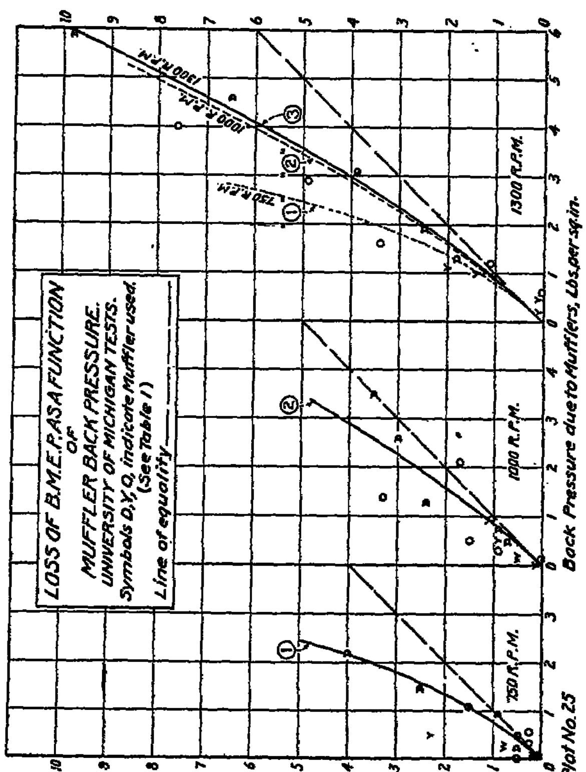
Plot No. 26

Muffler *Y*, besides giving the least back pressure and power loss, was the least in weight and size, and the best in silencing ability.

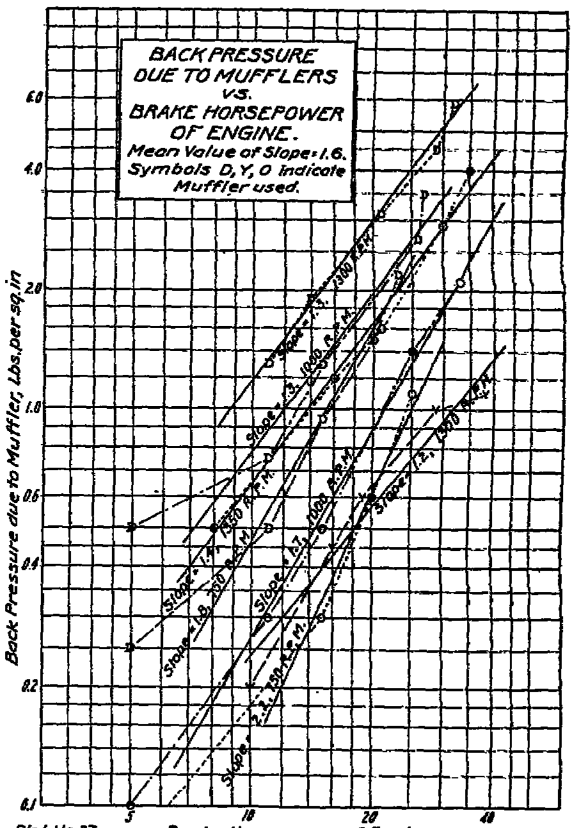
A partial analysis of the University of Michigan data is given in plot 25, showing the losses of brake M. E. P. as a function of back pressure at various speeds. At the right of plot 25 the curves for the three speeds are combined. The resultant grouping is comparable with Table II and plot 11 of the main body of this report. (Note that in plot 11 back pressures are in inches of mercury and in plot 25 in pounds per square inch.) The conclusions there reached are confirmed, at least qualitatively, from this independent source.

Plot 26 shows the back pressures due to the mufflers as a function of power output of the engine. It is similar to plots 7 and 12 of the main report. Plot 27 presents the same data in logarithmic plotting, just as plot 8 reproduces plot 7. The suggestion that back pressure varies as about the 1.5 power of the output of the engine is checked.

It does not seem quite true, however, that back pressure depends, for any one muffler and engine, solely on power output. The back pressure (and per cent power loss) is higher when the power is obtained by small throttle opening and high speed than by large throttle opening and low speed. It should be remembered, however, that in the discussion of muffler capacity we have pointed out that some mufflers in our own tests did not increase back pressure when the number of cylinders exhausting into the muffler was doubled. Other mufflers did increase



Plot No. 25



Plot No. 27.

back pressure under the same circumstances by 50 to 100 per cent; none quadrupled, as might be expected from ordinary flow laws. It seems probable that the differences between the curves for muffler *D* in plot 26 for speeds of 1,300, 1,000, and 750 r. p. m. are largely tied up with its having been a poor muffler design (in respect of power loss) and that such a difference would show much less with the better mufflers of our later tests.

O